HYBRID SPLIT AIR CONDITIONING WITH EVAPORATIVE CONDENSER

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HYBRID SPLIT AIR CONDITIONING WITH EVAPORATIVE CONDENSER

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CERTIFICATE

It is certified that the work contained in the thesis entitled **Hybrid Split Air Conditioning with Evaporative Condenser,** by **Y.Narendra,** has been carried out under my supervision and that work has not been submitted elsewhere for a degree.

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I I T Kanpur

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Nomenclature

A_j	Area walls and roof, $\left[m^2\right]$
C_{th}	Thermal capacity of wall, $[KJ]$
c	Specific heat capacity of building materials, $[KJ/KgK]$
C_1, C_2	Constant
cp	Specific heat capacity of refrigerant, $[KJ/KgK]$
DBT_{out}	Dry bulb temperature, $[°C]$
WBT_{in}	Wet bulb temperature, $[°C]$
d	Declination angle
h	Hour angle
h_o	Outside convective heat transfer coefficient, $\left[KW/m^2 / K\right]$
h_i	Inside convective heat transfer coefficient, $\left[KW/m^2 / K\right]$
h_f	Enthalpy of the fluid, $[kJ/kg]$

h_g Enthalpy of the vapour, $[kJ/kg]$

IST Indian Standard Time

I_{df} Diffused sky radiation, $[KW/m^2]$

I_{dn} Direct normal radiation, $[KW/m^2]$

I_{dr} Direct solar radiation, $[KW/m^2]$

I_r Solar radiation reflected from the surrounding surface

I_t Total radiation, $[KW/m^2]$

K_i Thermal conductivity of building materials, $[KW/m^2]$

LCT Local civil time

LST Local solar time

l Latitude

m_{ref} Mass flowrate of the refrigerant

N_{ach} Number of air changes

p Saturation pressure of a refrigerant, bar

$\dot{Q}_s$ Total solar load, $[KWh]$

$\dot{Q}_{\inf il}$ Total infiltration load, $[KWh]$

$\dot{Q}_{equip}$ Total equipment load, $[KWh]$

$\dot{Q}_{total}$	Total cooling load, $[KWh]$
q	Heat flux, $[KW/m^2]$
s_f	Entropy of the fluid, $[kJ/kg/K]$
t	temperature of refrigerant, $[°C]$
T_{max}	Maximum air temperature of a day, $[°C]$
T_{min}	Minimum air temperature of a day, $[°C]$
T_o	Outside air temperature, $[°C]$
$T_{w,o}$	Outside wall temperature, $[°C]$
T_i	Inside air temperature, $[°C]$
T_{sol}	Sol-air temperature, $[°C]$
U_j	Overall heat transfer coefficient, $[KW/m^2]$
V_{room}	Volume of room, $[m^3]$
v_i	Velocity of air inside the room, $[m/s]$
v_o	Velocity of air outside the room, $[m/s]$
V_g	Specific volume of dry and saturated refrigerant vapour, $[m^3/kg]$
[illegible]	Specific volume of superheated refrigerant vapour, $[m^3/kg]$
	Wet-bulb depression, $[°C]$

Δx_i	Thickness of wall, $[m]$

Greek letters

α	Wall solar azimuth angle
β	Solar altitude
γ	Solar azimuth angle
ε	Emissivity of the surface
η_{se}	Saturation efficiency
θ	Incidence angle
ϕ	Surface tilt angle from vertical
η_c	efficiency of the compressor
ϕ_1	Time-lag factor
ρ	Density of building material, $[KW/m^3]$
λ	Decrement factor

Subscript

i	Inside
o	Outside
se	Saturation efficiency
w	Wall
max	Maximum
min	Minimum

Abstract

The present work deals with hybrid air-conditioning system, on the basis of sacrifice of comfort by small fraction to achieve energy conservation drastically. This is emphasized in order to utilize the evaporative cooling and conventional air-condition as per available environmental conditions. The shortage of energy and escalating energy costs and environmental constraints have placed more emphasis on the design and simulation of thermal environmental systems.

A generalized programme is developed to calculate the solar heat gain and total cooling load for any building at any location. A programme is also developed to calculate the length of the capillary required for different pressure drops for given flow rate of the refrigerant and diameter of the capillary tube, for refrigerant R22.

Hybrid air-conditioning system consisting of Split Air-Conditioner and desert cooler is designed and fabricated to study energy saving. Desert cooler with semi elliptical pad is used. The conventional Split Air-Conditioner is modified and an additional evaporative condenser is incorporated.

Experiments were conducted in the laboratory. The split air-conditioner with evaporative condenser has given 22.3% increases in the refrigerating effect than the split air-conditioner with air-cooled condenser. The energy consumption (per ton of refrigeration) of split air-conditioner with evaporative condenser is 15.1% less than that with air-cooled condenser. For year round operation the hybrid air-conditioner gives 38.9% energy saving than that of the air conditioner with air-cooled condenser and 56.3% as compared to the air conditioner with evaporative condenser.

Chapter 1

Introduction

As the present work is dealing with hybrid air-conditioning system, it has culminated from the same philosophy. In this the emphasis has been stressed on the sacrifice of comfort to small fraction to achieve energy conservation drastically. By this concept it became possible to utilize the evaporative cooling and conventional air-condition as per available environmental conditions. A comfortable and healthful environment is now considered a necessity, and many modern processed products would not exist without precise control of environmental conditions. Therefore almost all homes, offices, and industrial structures are now designed with a means to control the indoor conditions throughout the year. In our modern society, however, the maintenance of cool environment during the warm months proves to be as important for the utilization of human resources and productivity with quality. The shortage of energy and escalating energy costs and environmental constraints however are causing reexamination of comfort conditions on one hand as essential and on the other hand the conservation of energy. All these considerations have placed more emphasis on the design and simulation of thermal environmental systems.

1.1 Description

Since human body can adapt to only certain limit itself to the changing conditions of the environment, it has become essential for human to provide an artificial environment. In their urge to provide the necessary condition for the human body the concept of air-conditioning has sprouted. Air-conditioning can be broadly divided into summer air-conditioning and winter air-conditioning. The conventional summer air conditioning uses a refrigeration system and a dehumidifier against a heat pump and a humidifier used for winter air conditioning. By and large, the term air conditioning in Indian context chiefly refers to the control of temperature in a down-ward direction i.e., when the hot air in summer can be cooled and dehumidified. Thus the concept of summer air-conditioning is most prevalent in our country.

The comfort air-conditioning for India and the process requirements for different climatic zones reveal that the most populous and important climatic region is monsoon type with dry winters. Generally, the present practice in the region is to use conventional air-conditioners throughout the year. In view of the energy conservation it is emphasized that the use of hybrid air-conditioning system will prove more beneficial. One can use conventional air-conditioning during the months July to September when humidity is high. A desert cooler can be used for the months April to June and October when the humidity is low. During winter the normal practice is to use electrical heaters which are most convenient method to serve the purpose, the use of heat pump will be more beneficial in the point of energy conservation.

1.2 Literature Survey

1.2.1Hybrid Air Conditioning

Due to global energy crisis and a need for energy conservation, Ramamoorthy [16] has reported a compromise between energy requirement and comfort level. Inside design conditions $T_{db} = 30°C$ and $\phi = 60\%$ having air velocity of 0.7 m/s or higher has been suggested as compared to commonly used near static air velocity of 0.13m/s for the comfort environment.

An extensive survey has been carried out in order to obtain the effective temperatures for hot humid and hot dry climatic conditions, [17]. The higher effective temperature becomes acceptable, as energy required to cause circulation of air is far less than that required for lowering the temperature.

Tanabe, S., And Jimura, S., [18] has carried an extensive research for elevating comfort condition. They have carried out extensive studies on human comfort for the hot dry (summer season) and hot humid (rainy season) climates in the light of " Energy Conservation".

1.2.2 Structure Heat Gain

The analysis of cooling load involves parameters such as the ambient air temperature, direct and diffused solar radiations, air velocity, characteristics of enclosing walls and the orientation of the building.

Threlkeld [3] has extensively described the process of heat transfer due to temperature difference between the external and internal environments. The periodic heat transfer through the walls has been explicitly dealt with, the using the concept of

sol-air temperature. His analysis essentially deals with heat transfer through homogeneous walls.

Mackey and Wright [4] have described in detail the periodic heat transfer through composite walls or roofs by reducing a composite wall problem into an equivalent homogeneous wall. The following three methods account for the periodicity of external conditions:

1. Threlkeld's classical approach [1]
2. Transfer function method [3]
3. Finite difference method [4]

Threlkeld's classical approach involves the concept of sol-air temperature and is used in the present work.

In the transfer function method the various components of space heat gain are added together to get an instantaneous total rate of space heat gain. It is then converted into an instantaneous space-cooling load through the use of weighting factors called coefficients of '*room transfer function*'. The transfer function is nothing more than a set of coefficients that relate to an output function at a given time to the value of one or more driving functions at the time of previous times.

Kadambi and Hutchinson [4] have described an approximate technique to determine one dimensional transient heat transfer through walls and roofs, in the form of the Finite Difference Method. The basic simplicity of approximate method contrasted with analytical techniques is asserted in their work.

A detailed procedure of cooling load calculations, from the point of view of practical designs, is outlined in ASHRAE Handbook of Fundamentals [10]. The

methodology and equations for hour-by-hour load calculations are presented in the ASHRAE handbook of fundamentals.

1.2.3 Evaporative Cooling

The increasing cost of electric energy has necessitated an economic method of cooling. The evaporative cooling is one of such method. Leonardo da Vinci built water powered evaporative cooler for the bedroom of his patron's wife for first time. It gave reasonable comfort conditions inside the room. The first mechanical direct evaporative cooler was developed in about **1932**.

The characteristics of dry tropical climate are given in Jannot Yves [15]. The needs of economic equipment for air-cooling are outlined from climatic data and local financial possibilities. For this climatic zone, an acceptable thermal comfort can be maintained all the year round by direct evaporative cooling of outside air. Experimental results obtained by testing a direct evaporative cooler built locally at a compatible cost are presented.

1.3 Present Study

A generalized programme is developed to calculate the solar load and total cooling load for any building at any location. A programme is developed to calculate the length of the capillary required for different pressure drops for any assigned flow rate of the refrigerant and diameter of the capillary tube. A programme is developed for calculating the properties of R22.

Hybrid air-conditioning system consisting of Split Air-Conditioner and desert cooler is designed and fabricated to study energy saving. Desert cooler with semi elliptical pad is used. The conventional Split Air-Conditioner is modified and an

additional evaporative condenser is fitted. Taking the reference of Mr. Raj Kumar's Ph.D. work, the evaporative condenser was fabricated.

Experiments are conducted in the laboratory. The split air-conditioner with evaporative condenser has given 22.3% increases in the refrigerating effect than the split air-conditioner with air-cooled condenser. The energy consumption (per ton of refrigeration) of split air-conditioner with evaporative condenser is 15.1% less than that with air-cooled condenser. For year round operation the hybrid air-conditioner gives 38.9% energy saving than that of the air conditioner with air-cooled condenser and 56.3% as compared to the air conditioner with evaporative condenser.

Chapter 2

Evaporative Cooling

Introduction

Evaporative air-cooling occurs in nature near waterfalls, flowing streams over lakes and oceans, under summer showers and even upon wet skin. The classic example of evaporative cooling which we read in elementary school science books is the cooling effect felt, when a moistened hand is waved in the air. The evaporative process simply removes sensible heat (i.e., cooling by decreasing the surface temperature) and replaces it with latent heat (i.e., increases the moisture content of air)[11,12]

2.1 Primitive Evaporative Cooling

Evaporative cooling was probably one of the first mechanical cooling methods used by men. Egyptian painting from 2500 B.C. shows slaves fanning porous clay jars to get chilled water. A wall painting in Herculaneum of 70 A.D. shows a leather water bottle used for cooling water by similar means. Both the American Indians of the South West and the ancient Persian cooled their tents by

damp felt or grass mats. Leonardo Da Vinci built water powered evaporative cooler for the bedroom of his patron's wife.

In the ancient period in India, evaporative cooling was even used to make ice. For this shallow beds dug in the earth was filled with 300mm straw, upon which shallow earthen pans were placed. For still frosty nights, even though air temperature never fall less than 6°C, ice would form sometimes 40 mm thick due to adequate refrigeration produced by evaporation and radiation in the night sky. The people here still use the ancient cocos tatti with greater effectiveness. Doors and windows were kept wet traditionally by coolies, now replaced by pumps with catch basin.

2.2 Modern Evaporative Cooling

In modern days, evaporative coolers are extensively used in textile mills, industrial plants and homes after 1900 A.D. Textile mills were natural applicants as they need both cooling for comfort and high humidity to keep the fibers workable. By 1953, evaporative cooling had become a $30 million per year industry, providing people with the means to cope with the summer climate. Before automobile air-conditioning was available, small evaporative coolers were used in the Southwest to cool cars. Even today in India people use khuss-khuss tatties on the roof of the car for evaporative cooling.

The first mechanical direct evaporative cooler was developed in 1932 when householders assisted their wet-cloth hung windows with electric fans. By 1933, thousands of homemade coolers had appeared in Arizona. At first, these were burlap covered wooden frames wet by dripping water and mounted at outside windows. In 1934, excelsior pads replaced burlaps.

Today evaporative coolers are commonly used in residential and commercial buildings in tropical region with limited commercial and industrial application

elsewhere. There are also applications of evaporative cooling in large commercial HVAC systems.

2.3 Theory

Evaporation is described as an adiabatic process, meaning that the total amount of heat in the thermal system remains constant. As water evaporates, the sensible heat content of the system falls, while the latent heat content increase by an equal amount. In other words, dry-bulb temperature falls, but moisture content raises. The limit of temperature reduction is up to the air wet-bulb temperature at the beginning of the process. Surface evaporation can cool up to the wet-bulb temperature. The process stops when the relative humidity approaches 100%. A simple measure of the potential for evaporative cooling at any given air condition is the wet-bulb depression, or the difference between the dry-bulb and wet-bulb temperature. It provides an upper limit of the achievable temperature change by direct evaporation.

The evaporative cooling results from a material phase change, i.e., in particular evaporation of water. The amount of heat absorbed by a material as it changes from liquid to vapour is called the latent heat of vaporization. For water, the latent heat of vaporization is given approximately by:

[2500-2.3 x (water temperature in K-273.15)] KJ/kg

the initial temperature of the water being evaporated has some effect on the net heat absorbed. However, for most processes, it can be assumed that each kilogram of water that evaporated absorbs 2442.5 kJ of heat, corresponding to water temperature of 298K.

Fig 2.1 shows the evaporative cooling process of laboratory made evaporative cooler as a part of present study for September condition at Kanpur. The outdoor condition (304.2K dry-bulb temperature and 76% R.H.) is represented by point A on the psychometric chart. The condition of air inside the building is at 301K and 86% R.H., represented by C. Outside air is sent through the evaporative system and leaves it at saturation condition B. This idealized process occurs along a constant enthalpy and stops at 100 % R.H. The final indoor air condition will lie some where along the

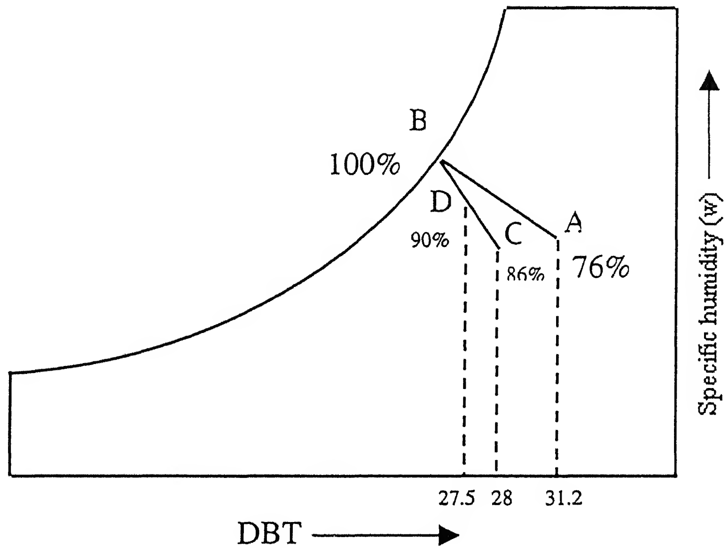


Figure 2.1: Direct Evaporative Cooling

line from B to C. its exact location depends on the relative masses of the two air streams. A mixture of one half evaporatively cooled air and one half outside air would provide air at 300.5°C and 90% R.H. This is shown by point D in the figure.

2.3.1 Actual Evaporative Cooling Process

Actual evaporative cooling systems have inherent inefficiencies and limitation that prevent them form operating as ideal process. The idealized process may be considered as the absolute maximum attainable performance. The real system will always provide less cooling i.e., higher temperature.

In an ideal evaporative system, all the air sent through the system is completely saturated with moisture and leaves at 100% R.H. In actual situation all water is not evaporated and so air is not fully saturated. The capability of an evaporative cooler to cool and humidity the air is measured by *saturation efficiency* η_{se} as:

$$\eta_{se} = \frac{DBT_{out} - DBT_{in}}{WB_{depression}} X100 \qquad \textbf{(2.1)}$$

Commercially produced systems provide saturation efficiencies of about 80 %, with sometime as low as 50 % and other as high as 90 %. The high humidity is not desirable as it renders favorable state for development of bacteria. Hence, it provides unhealthy indoor condition. From this point of view about 65% humidity is the upper limit of humidity. Beyond this bacteria growth increases rapidly.

2.4 Evaporative Cooling Equipment

Evaporative cooling equipment may use either a direct or indirect cooling process. In the direct process the water is evaporated directly into the air stream that flows into the conditioned space. In an indirect cooler the evaporation process is separated from the air to be delivered to the conditioned space. Evaporative coolers may also be either single-stage or multi-stage. Multi-stage evaporative systems use

two evaporation processes in series, with the first supplying pre-cooled air to the second stage.

2.4.1 Direct Evaporative Cooling

In direct evaporative cooler water is supplied through a float valve to a small reservoir and then flows down through fibrous pads. A fan draws large volumes of outdoor air through pads, where it is cooled by evaporation, and then supplied to the building. This cool and more humid air absorbs sensible heat from the building.

The temperature of air delivered by an evaporative cooler may be estimated by the following equation:

$$T_{supply} = DBT_{out} - \left(WB_{depression} \times \eta_{se}\right) \quad (2.1)$$

or

$$T_{supply} = DBT_{out} - \left(DBT_{out} - WBT_{out}\right) \times \eta_{se} \quad (2.3)$$

Using the typical saturation efficiency, i.e., 60% this equation can be rewritten as

$$T_{supply} = 0.4DBT_{out} - 0.6WBT_{out} \quad (2.4)$$

this equation is useful for judging the applicability of direct evaporative coolers under different conditions. When conditions permit their use, evaporative coolers are much more economical method of cooling than conventional vapor-compression air-conditioners. The power consumption of fan ranges from 250 W to 750 W in most residential units. A comparable air-conditioner might consume six to eight times as much electric power. Water consumption in an evaporative cooler is quite significant. A typical evaporative cooler with a saturation efficiency of 80% requires about 9 liters of water per hour of airflow for each 6K of wet-bulb depression. An

evaporatively cooled building operates as an open system where dry out-door air is continuously drawn into cooler, conditioned, and supplied to the building, providing a complete air change every 2 to 3 minutes.

Direct evaporative coolers humidify the air supplied to a building, rendering the relative humidity of indoors to be always higher than that of outdoors. The successful application of direct evaporative coolers depends on the existence of outdoor humidity levels well below human comfort conditions. But even in arid zones, they are considered as second-class cooling system, because they cannot dehumidify consistently as done by conventional air-conditioner. Consequently, evaporative systems should be installed with conventional air conditioning systems. An air-conditioning *system which incorporates evaporating cooling and mechanical air-conditioning system* is called hybrid air-conditioner.

2.4.2 Indirect Evaporative Cooling

Indirect evaporative coolers attempt to make use of phase change process without increasing the amount of moisture in the supplied air to building. As such indirect coolers use a heat exchanger to separate the direct evaporative process from the air to be delivered to the building. The heat exchanger may be of tube and fin type, corrugated sheets of plate-to-plate heat exchanger. A direct evaporative process cools air that flows across one side of the heat exchanger, removing heat, and is then exhausted to the outdoors. The air to be supplied to the building flows across the other side of the heat exchanger and is cooled without receiving moisture directly.

The cooled water from the cooling tower is circulated through the heat exchanger by a small pump. A fan draws the air through the heat exchanger and supplies it to the building. The system is quite effectively in dry climates, often reducing indoor temperature by 10 to 20K on a hot and dry climate. A simple indirect cooling process is shown on the psychometric chart in Fig 2.2. Outdoor air at

314.5K dry-bulb temperature and 22.7% R.H. enters the cooler at point A and cooled without the addition or removal of moisture along a horizontal line to point B, 301K dry-bulb temperature and 57% R.H. (the actual output conditions depend on the saturation efficiency of the direct evaporation process and the effectiveness of the heat exchanger).

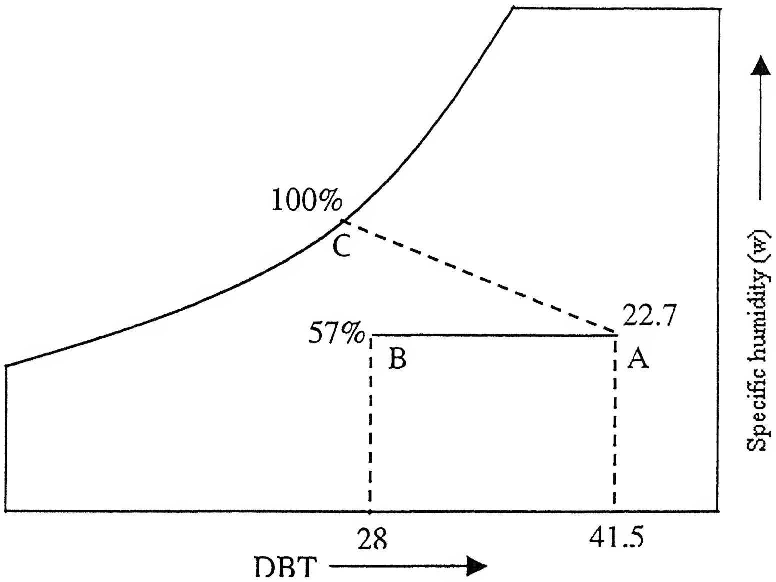


Figure 2.2: Indirect Evaporative Cooling

Two-stage evaporative cooling system offers significant improvements in performance over indirect evaporative cooking systems. In a two-stage evaporative system, the cooled air from an indirect cooler is passed through a direct evaporative cooler for additional cooling.

Chapter 3

Appraisal of Cooling Load

3.1 Different Heat Sources

The heat load arises from the following sources:

1. Energy transfer from the surroundings to the system. These are:

    - Heat transmission through barriers such as walls, doors, ceiling, floor, etc. being caused by the temperature difference existing on the two sides of the barrier.
    - The solar heat gain:

        Absorbed by walls or roofs exposed to radiation from the sun and transferred to the inside space.
    - Heat and moisture introduced to the conditioned space through infiltration and ventilation air.

2. Heat generation within the confined space. They are:

    - Heat from different equipment such as fans, lights and other electrical appliances in use.
    - Heat from occupants

3.2 Need For Hourly Cooling Load Calculations

The ambient air temperature varies during the day and its variation also depends upon the day of a month. Accordingly, the heat load varies over a period of 24 hours. This is mainly due to the variation in the solar intensity falling on the earth surface. The load calculation is further complicated by the fact that a wall has thermal capacity, due to which a certain amount of heat passing through it is stored and is transmitted to the inside at some time latter it also transfers heat to surroundings. Therefore, calculation based on instantaneous heat transmission through structure without considering the thermal capacity of the wall is erroneous. Moreover, the traditional methods of evaluating the cooling load involve various assumptions like the load from each component is constant in a day and maximum, which is not correct. Since the occurrence of maximum load at particular time is not the occurrence of the other components. It is proper to adopt an hourly cooling load calculation to facilitate an economic selection of a refrigeration system.

3.3 Heat Transfer through Structure

3.3.1 Building Survey

The building survey will include the flowing particulars:

- Location of the building, i.e., longitude and latitude.
- Orientation.
- Dimension of the building structure, such as length, breadth, height and thickness of each layer of the building material.
- Composition of building material and their physical properties.

3.3.2 Hourly outside Temperature Variation

Fig 3.1 shows the hourly variation in outside temperature for Kanpur. The meteorological data reveals that the minimum temperature occurs just before sunshine (may be one or two hours) while the maximum temperature will occurs three to four hours after the solar noon [1]. Since, the temperature time relation is available for only a few places an hourly variation in temperature is predicted base on maximum and minimum temperatures. It has been taken in the following form:

$$T = L + M\cos(15t - N)$$

Where, T= temperature at any time in °C.

L, M and N are constants, calculated by applying the boundary conditions:

$$\frac{dT}{dt} = 0, \text{ for } T = T_{\max} \text{ or } T_{\min} \qquad (3.1)$$

It is assumed that $T_{\min}$ occurs one hour before the sunshine and $T_{\max}$ occurs 12 hours thereafter.

The comparison of the temperature variation by Eq 3.1 with actual variation is also shown in Fig 3.1. The data has been taken from the average temperature of three years i.e., 1993,1994,1995 [8]

3.3.3 Solar Radiation

Solar radiation [9] forms the significant part of cooling load in air-conditioning. The total radiation intensity (I_t), reaching on terrestrial surface is the sum of the direct solar radiation (I_{dr}), the diffused sky radiation intensity (I_{df}) and the solar radiation intensity reflected from the surrounding surface (I_r). The

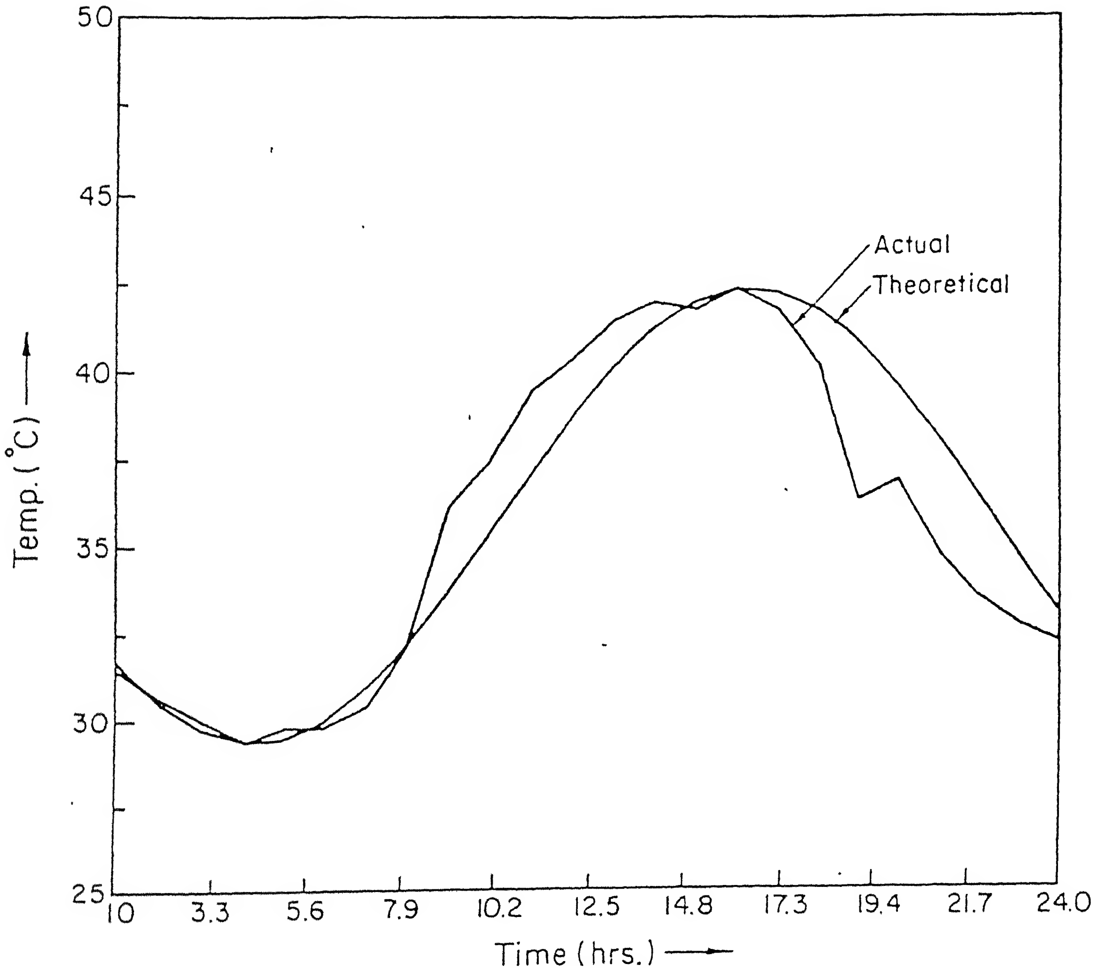


Figure 3.1 Hourly variation of actual and theoretical temperature.

intensity of the direct component is the product of the direct normal radiation intensity (I_{dn}) and the cosine of the incidence angle (θ) between the incoming solar rays and a line normal to the surface as shown in Figure 3.2.

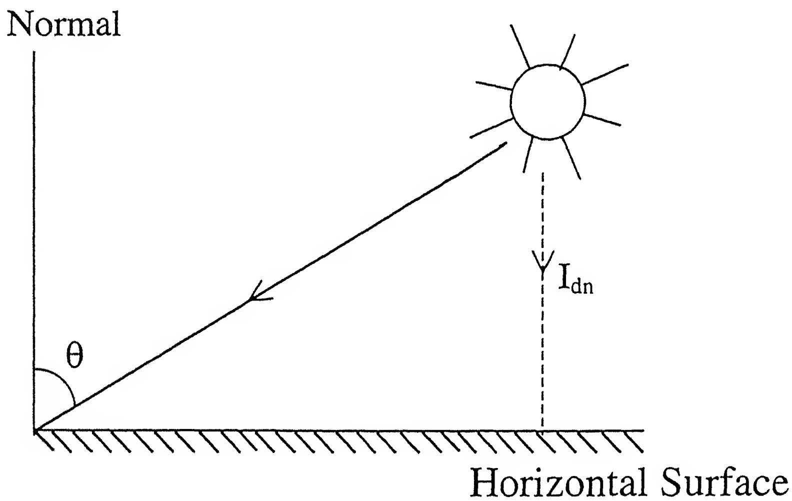


Figure 3.2. Direct Solar Radiation on Horizontal Surface

Thus,

$$I_t = I_{dn} \cos\theta + I_{df} + I_r \tag{3.2}$$

In the present analysis the reflected radiation (I_r) is neglected.

I_{dn} and I_{df} are given as:

$$I_{dn} = \frac{A}{\exp\left(B/\sin\beta\right)} \tag{3.3}$$

$$I_{df} = C \cdot I_{dn} \cdot F_{ss} \tag{3.4}$$

Value of $F_{ss} = 0.5$ for vertical surfaces, 1.0 for horizontal surfaces and $(1+\cos\phi)/2$ for any other inclined surfaces.

The values of A, B and C are tabulated in [9] and are given in Appendix '**B**'.

3.3.4 Determination of Incident Angles

The sun's position in sky is most conveniently expressed in terms of solar altitude β, above the horizontal and solar azimuth angle γ, measured from south. These angles β and γ depends on the latitude of the place *l*, solar declination *d*, and hour angle *h*, β and γ are related with *l*, *d*, and *h* by the following expression:

$$\sin\beta = \cos l.\cos d.\cosh + \sin d.\sin l \qquad (3.5)$$

and

$$\cos\gamma = \frac{\sin\beta.\sin l - \sin d}{\cos\beta.\cos l} \qquad (3.6)$$

The hour angle is calculated as follows:

$$\text{LST} = \text{LCT} + (\text{Equation of time}) \qquad (3.7)$$

Where, LST is local solar time and LCT is local civil time, which are based on the standard meridian of the country. And due to the irregularities of the earth's rotation, obliquity of earth's orbit and other factors, a solar day is not exactly equal to 24 hours. To counter that effect equation of time is added in LCT, which is given by

$$\text{LCT} = [\text{IST} - (82.5 - \text{longitude of location})]/15 \qquad (3.8)$$

Where IST is Indian Standard Time which is based on 82.5 meridian (standard longitude for India) passing through Naini near Allahabad.

The hour angle is zero at solar noon and increases by 15° every hour. Hence hour angle,

$h = (12 - \text{LST}) \times 15$, before solar noon (3.9)

$h = (12 + \text{LST}) \times 15$, after solar noon (3.10)

If a surface is tilted by an angle ϕ to the vertical position then the incidence angle θ is given by

$$\cos\theta = \cos\beta.\cos\alpha.\cos\phi + \sin\beta.\sin\phi \quad (3.11)$$

If the surface is vertical, $\phi = 0$, then

$$\cos\theta = \cos\beta.\cos\alpha \quad (3.12)$$

If the surface is horizontal, $\phi = \pi / 2$, then

$$\cos\theta = \sin\beta \quad (3.13)$$

After calculating value of h, the values of β and γ are calculated. The wall solar azimuth angle, α [9] for various walls are:

$$\alpha_E = \begin{pmatrix} |\pi / 2 - \gamma|, \textit{beforenoon} \\ |\pi / 2 + \gamma|, \textit{afternoon} \end{pmatrix} \text{ east facing wall} \quad (3.14)$$

$$\alpha_W = \begin{pmatrix} |\pi / 2 - \gamma|, \textit{beforenoon} \\ |\pi / 2 + \gamma|, \textit{afternoon} \end{pmatrix} \text{ west facing wall} \quad (3.15)$$

$$\alpha_N = \begin{pmatrix} |\pi - \gamma|, \textit{beforenoon} \\ |\pi + \gamma|, \textit{afternoon} \end{pmatrix} \text{ north facing wall} \quad (3.16)$$

$\alpha_S = \gamma$, south facing wall (3.17)

3.3.5 Sol-Air Temperature

The calculation of space cooling load as a result of heat gain through exterior roof and walls involves the concept of sol-air temperature [1]. A heat balance at a sunlit surface gives the heat flux to the surface as:

$$q = \alpha_\tau I_t + h_o(T_o - T_{w,o}) - \varepsilon\Delta R \qquad (3.18)$$

$\varepsilon=1$, for black body

$\Delta R = 0.063\, kW/m^2$, for horizontal surface, 0 for vertical surface [9].

The above equation can also be written as

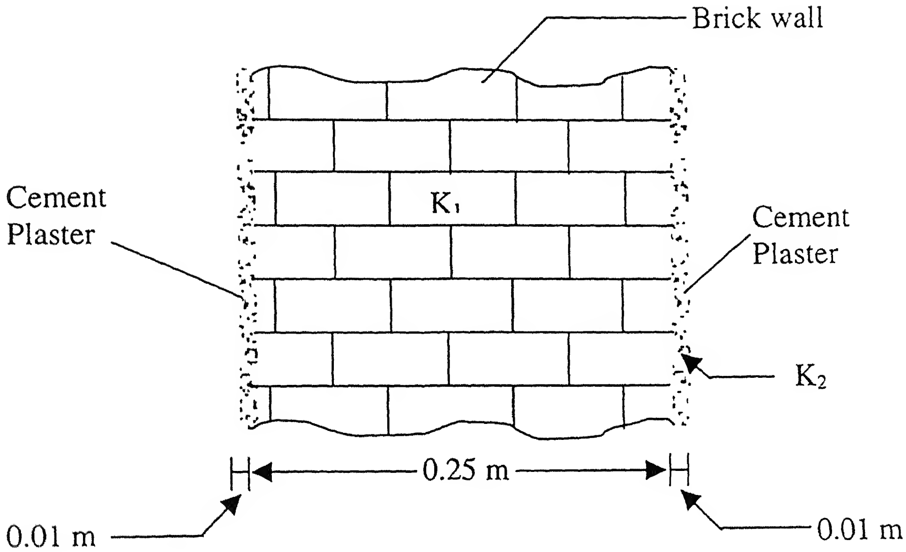
$$q = h_o(T_{sol} - T_{w,o}) \qquad (3.19)$$

Where,

$$T_{sol} = T_o + \alpha_\tau I_t / h_o - \varepsilon\Delta R / h_o$$

3.3.6 Heat Transfer through Walls and Roof Using Decrement Factor and Time Lag

As building materials have finite thermal capacity, the instantaneous heat transfer through walls and roofs by neglecting thermal capacity of the building materials is not correct. The structural detail of walls and ceiling is given in Fig 3.3.



K1= 0.00132 kW/m K

K2= 0.00865 kW/m K

Figure 3.3 Structural Detail of Ceiling

Thermal capacity of wall C_{th}, is expressed as:

$$C_{th} = mc = \rho c V = \rho c A t \tag{3.20}$$

Due to thermal capacity of a structure the following two effects have been observed [7]:

1.There is a time lag between the heat transfer at the outside surface and the inside surface.

2. There is decrement in heat transfer due to the absorption of heat by the wall and subsequent transfer of a part of this heat back to the outside air when temperature of ambient is lower.

Considering these two facts, actual heat transfer at any time 't' is given by:

$$\dot{Q}_{\tau} = \sum_{j=1}^{5} \left[U_j A_j (T_{sol} - T_i) + A_j U_j \lambda_j (T_{o(\tau-\varphi)} - T_{sol})\right] \quad (3.21)$$

The values of ϕ and λ with respect to wall thickness are given in Figs 3.4 and 3.5 respectively. In the present study Eq 3.21 is used to calculate the heat transfer through the structures.

3.3.7 Over-all heat transfer coefficient for walls and roof

To get overall heat-transfer coefficient U, the value of h_o and h_i are taken from [4], as given below:

For exterior wall

$$h_o = 0.00391(0.399 + v_w) \quad (3.22)$$

For roof

$$h_o = 0.00141(1.454 + v_w) \quad (3.23)$$

For the inner surfaces of the enclosed space, the convective heat transfer coefficient is found to be the function of the inside air velocity and the temperature difference existing between the room air and inner surface of the wall. In terms of interior air velocity v_I and temperature difference ΔT_I, we get

For interior walls

$$h_i = c_1(\Delta T_i)^{0.25} + 0.00391(0.399 + v_i) \quad (3.24)$$

For ceiling

$$h_i = c_2(\Delta T_i)^{0.25} + 0.00141(1.454 + v_i) \quad (3.25)$$

Where, c_1 and c_2 are constants and its values are

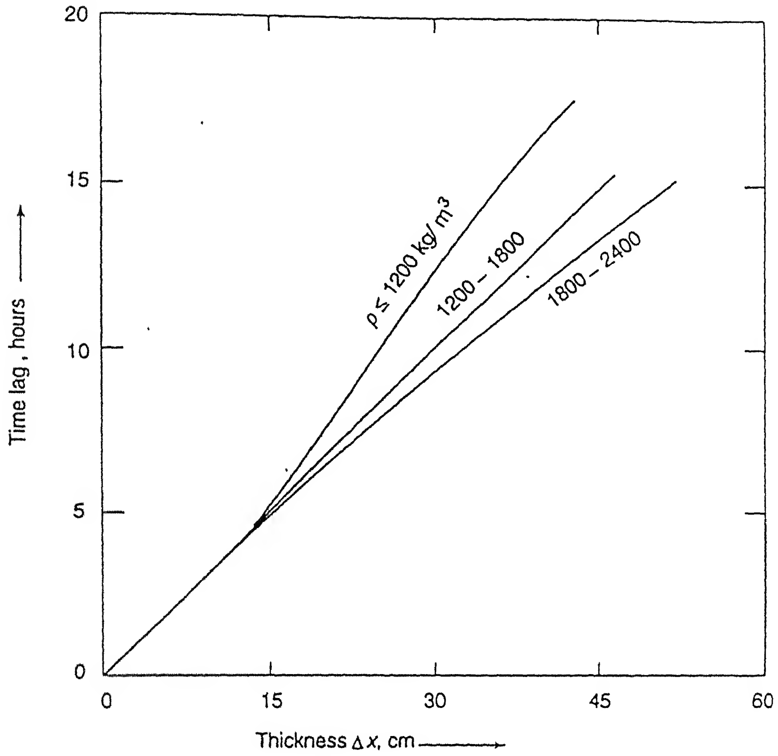


Figure 3.4 Variation of time-lag with wall thickness.

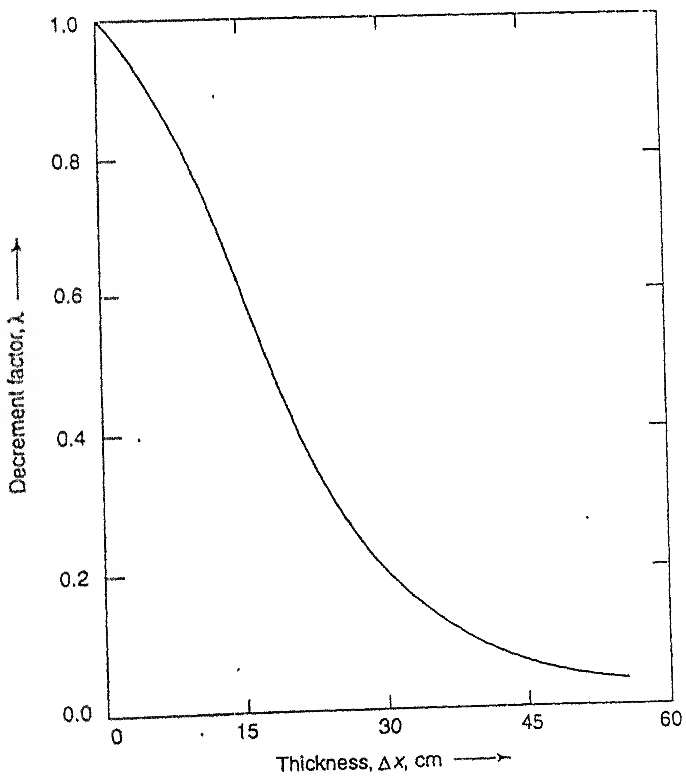


Figure 3.5 Variation of decrement factor with wall thickness.

$$c_1 = 1.25X10^{-3}, c_2 = 1.45X10^{-3}$$

Now, the value of U for walls and roof taking into account of the time-lag is calculated as follows:

$$q = U\left[(T_{sol} - T_i) + \lambda(T_{o(t-\phi)} - T_{em})\right] = UD \tag{3.26}$$

Where,

$$D = \left[(T_{sol} - T_i) + \lambda(T_{o(t-\phi)} - T_{em})\right] \tag{3.27}$$

For the interior wall:

$$q = \left[c_1(\Delta T_i)^{0.25} + 0.00391(0.399 + v_i)\right]\Delta T_i \tag{3.28}$$

$$q = c_1(\Delta T_i)^{1.25} + h_w \Delta T_i \tag{3.29}$$

Where,

$$h_w = 0.00391(0.399 + v_i)$$

Comparing the Eqs. 3.26 and 3.28 we get

$$c_i(\Delta T_i)^{1.25} + h_w \Delta T_i = UD \tag{3.30}$$

But

$$U = \frac{1}{\frac{1}{h_o} + \sum \frac{\Delta x_i}{k_i} + \frac{1}{h_i}} \tag{3.31}$$

Putting

$$\sum \frac{\Delta x_i}{k_i} + \frac{1}{h_o} = R = constant \tag{3.32}$$

for a given wall and for constant outside wind velocity (v_w) Eq 3.32 becomes

$$c_1(\Delta T_i)^{1.25} + h_w \Delta T_i = \frac{D}{\frac{1}{h_i} + R} = \frac{D}{\frac{1}{c_1 \Delta T_i^{0.25}} + R} \quad (3.33)$$

$$D = \Delta T_i + R\, c_1(\Delta T_i)^{1.25} + R\, h_w \Delta T_i \quad (3.34)$$

the newton-Raphson iterative procedure has been used to get ΔT_i.

Accordingly we have

$$F = \Delta T_i + R\, c_1(\Delta T_i)^{1.25} + R\, h_w \Delta T_i - D \quad (3.35)$$

Thus, after getting value of ΔT_i, h_i is calculated and hence values of U's for different walls and roof is calculated.

3.4 Infiltration Load

There is always leakage of outdoor air into a building through cracks and openings caused by pressure difference across the boundary surfaces. This leakage is called infiltration load. The exchange of air may cause both types of load namely latent heat load and sensible heat load. The load due to infiltration is given by the following expression:

$$\dot{Q}_{infil} = \frac{V_{room} N_{ach} (H_{o,a} - H_{i,a})}{v_{air}(24X3600)} \quad (3.36)$$

3.5 Lighting and Other Electrical Appliances

In general the instantaneous rate of heat gain from electric lighting is calculate from

$$\dot{Q}_{light} = \text{Total Light Wattage x Cooling Load Factor (CLF)}$$

CLF is a function of time of use, type of arrangement, room furnishing etc. in present study, a value of 0.88 is taken from [9].

Other electrical appliance used normally is fan and the instantaneous load from the fan is equal to the total rating $\left(\dot{Q}_{fan}\right)$

Therefore,

$$\dot{Q}_{equip} = \dot{Q}_{light} + \dot{Q}_{fan} \tag{3.37}$$

3.6.Total Load

Total load is the sum of all the loads, i.e., Solar load, infiltration load, equipment load.

$$\dot{Q}_{total} = \dot{Q}_{solar} + \dot{Q}_{\inf il} + \dot{Q}_{equip} \tag{3.38}$$

Chapter 4

Split Air Conditioners

Introduction

A split a conditioner is a modified form of window air conditioner in which the condensing unit is separated from the evaporator unit. This facilitates the condensing unit to be placed in a remote place from the room that is to be air-conditioned.

In the recent years split air-conditioners have gained popularity because of the following merits:

- It is a substitute for air-conditioning partition rooms where window air-conditioners can't be used
- It is silent in operation because of the remoteness of the compressor
- The room side unit can be of the customer choice to match the interior decoration of the room

4.1 Types of Split Air-Conditioners

a. Direct room mounted split unit

The evaporator of this type can be installed on floor, wall or ceiling. The condensing unit is kept remote in suitable location. Our present set-up is of this type. Fig 4.1 shows the layout of the direct room mounted split air conditioner.

b. Ductable split unit

In this method the evaporator is concealed and normally mounted above false ceiling space and the cold air is supplied through ducting and delivered through the terminals located at selected places.

c. Multi split units

This system offers the feature of having individual room temperature control.

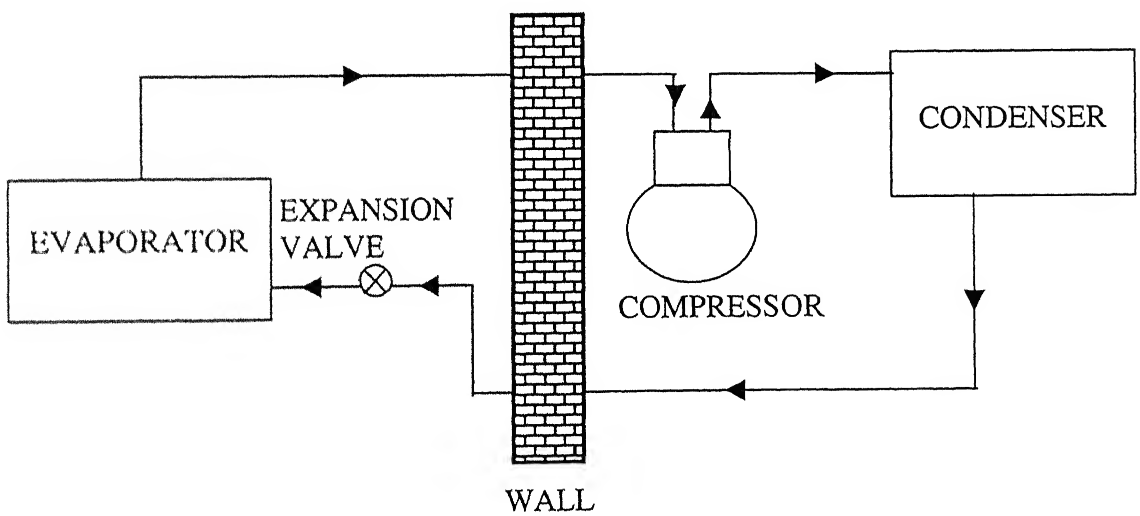


Figure 4.1. Line diagram of split air-conditioner.

4.2 Installation

In split air-conditioners the evaporator and condensing units are connected by refrigerant tubing. The condensing unit can be mounted at a higher orientation than the evaporator unit or below or at the same level as the evaporator unit.

The condensing unit in each site should be kept as near as possible to minimize the pressure drop in the connecting tubing, bends etc. Similarly mounting of condensing unit at a higher level than evaporator unit shall be preferably be avoided, if other options are available, to make the oil return to compressor easily.

The following precautions should be taken in installing the split air-conditioner to ensure reliable performance of the unit:

4.2.1 Evaporator

- The external surfaces of the evaporator coil is cleaned with detergent water, vaniklin etc
- Number of fins in the coil and the fin bonding over the tube at different places is properly inspected
- Blower is checked for noise and vibration
- Bigger water drain hole must be provide in the unit to avoid any restriction in the water drain
- Between capillary and evaporator a tube of 4.76mm OD and approximately 0.254m long is connected to minimize the hissing noise in the evaporator coil.

4.2.2Condenser

- The fin bonding of condenser coil is checked for good condition

- The fan motor is checked for vibration and touching of any sheet metal by fan blades
- To protect the unit from direct sun rays, an overhead sun shade must be provided
- Re-circulation of hot air either due to wind direction or an adjacent unit must be prevented

4.2.3 Refrigerant tubing layout

- Liquid refrigerant from entering the compressor under both operating and idle conditions is avoided
- Positive oil return to the compressor without oil getting trapped in any part of the system
- Minimum number of elbows used to minimize the pressure drop
- Correct line size must be maintained for optimum operating efficiency
- Suction line should be well insulated
- Liquid line does not require insulation. Suction-liquid heat exchangers shall not be used in split air-conditioners

4.2.4 Ventilation for room

The split air-conditioners mounted directly inside the room normally do not have any in-built provision to supply fresh outside air for ventilation of the room. In certain applications while using split air-conditioners, adequate quantity of fresh outside air has to be supplied based on judicious consideration and for this suitable external provisions may be made.

Chapter 5

Experimental Set-Up and Fabrication

Objective

Survey of the environmental conditions based on meteorological data available shows that the air has low humidity during April to June and October. When the comfort region is plotted on psychometric chart, it is seen that during above months there is considerable scope of evaporative cooling for reasonable level of comfort [19]. However, during July to September, humidity is very high. Hence it is common practice to adopt the conventional mechanical air conditioning system. Here it is iterated that the conditions during April to June and October are suitable for evaporative cooling. But due to natural changes in the environment, evaporative cooling may not render desirable level of comfort. As such under such situation the hybrid air-conditioning system may be operated under the conventional air-conditioning mode to maintain the same spirit of comfort. As the present system is capable of operating in either mode at any time, under such situation, the comfort can be achieved by conventional air conditioning mode of operation.

5.1 System Description

The present system is hybrid air-conditioning system with split air-conditioner with air-cooled condenser and evaporative condenser. In this system conventional split air-conditioner is modified with evaporative condenser to decrease the condenser pressure. The performance of this system is discussed in chapter 6. The schematic diagram of the experimental set-up is shown in plate 1. Taking the reference of Mr. Raj Kumar's Ph.D. work, the evaporative condenser was fabricated. The various components of the system and their description are discussed as below [20]:

5.1.1 Conventional Split Air-Conditioner

(a) Compressor

Model	: AH5522E
Rated Capacity	: 1.5 TR
Displacement Volume per rev.	: 39.82 cc.
Rated Power	: 2000 W
Rated Current, LRA	: 9.1 Amps, 45 Amps
Operating Voltage Range	: 180-260 V
Weight	: 30.3 kg

(b) Condenser

Cell size	: 26"x22"
Tube size	: 3/8"
Coil	: 27"x22 pipes, 2 rows
Fan motor	: 930 RPM, 1/8 hp

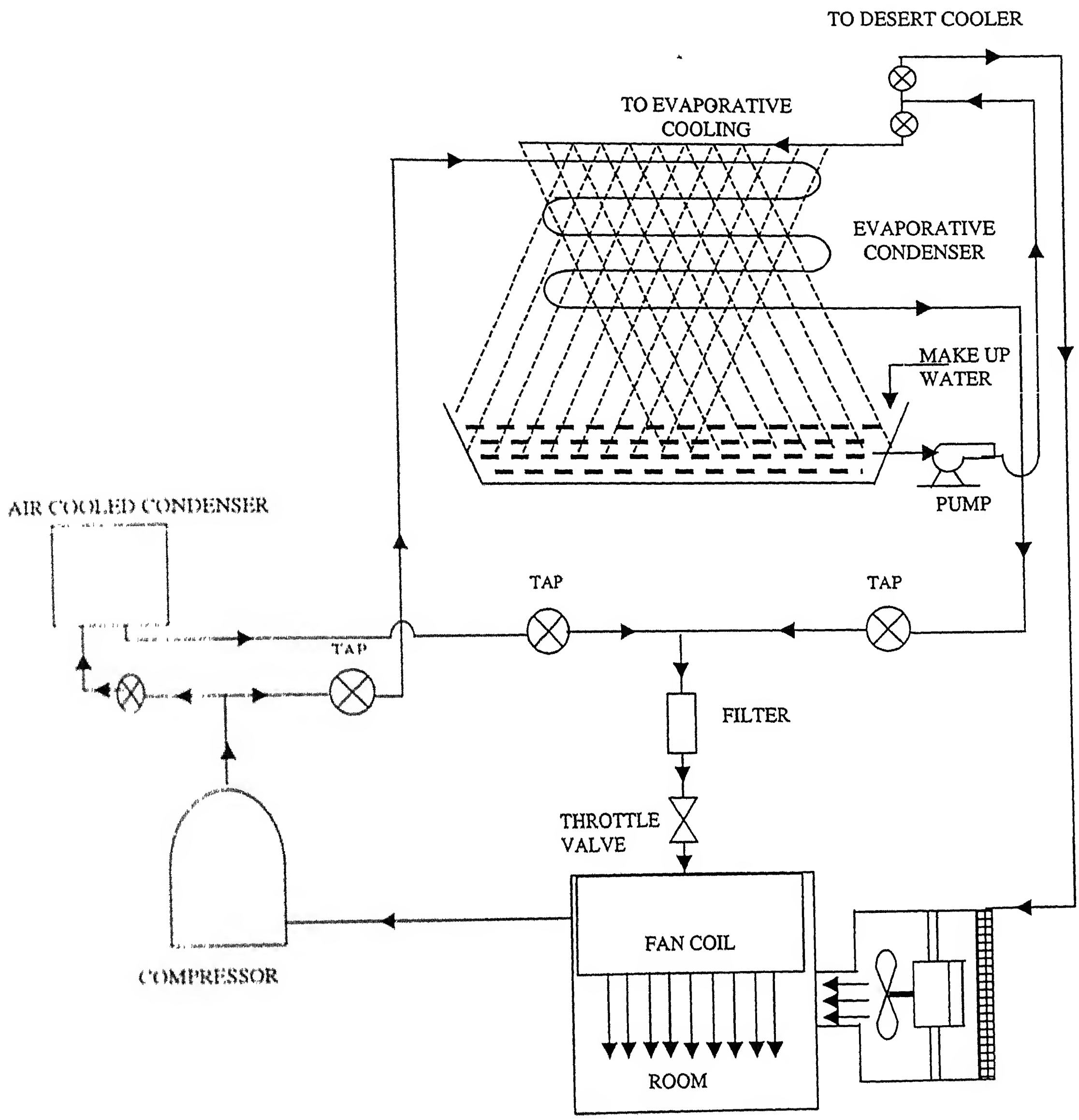


PLATE 1. SCHEMATIC DIAGRAM OF THE EXPERIMENTAL SET-UP

(c) Evaporator

Coil	: 65"x10 pipes, 3 rows
Pipe size	: 3/8"
Blower motor	: 1/5 hp

(d) Capillary

Diameter	: ID1.4 mm
Length	: 2 Nos., 9" each

5.1.2 Split Air-Conditioner with Evaporative Condenser

(a) Condenser

Cell size	: 24"x21"
Tube size	: 3/8"
Coil	: 21"x22 pipes, 5 rows

(b) Pump

Motor	: 2800 RPM, 150 W
Total head	: 13.5 m

(c) Capillary

Diameter	: ID1.7mm
Length	: 2 Nos. 11" each

The remaining components are the same as conventional split air conditioner.

The photo1 shows the evaporator unit of the split air-conditioner placed inside the room. The photo2 shows the conventional air-cooled condenser unit installed outside the room. The photo3 shows the evaporative condenser. And the photo4 shows the desert cooler.

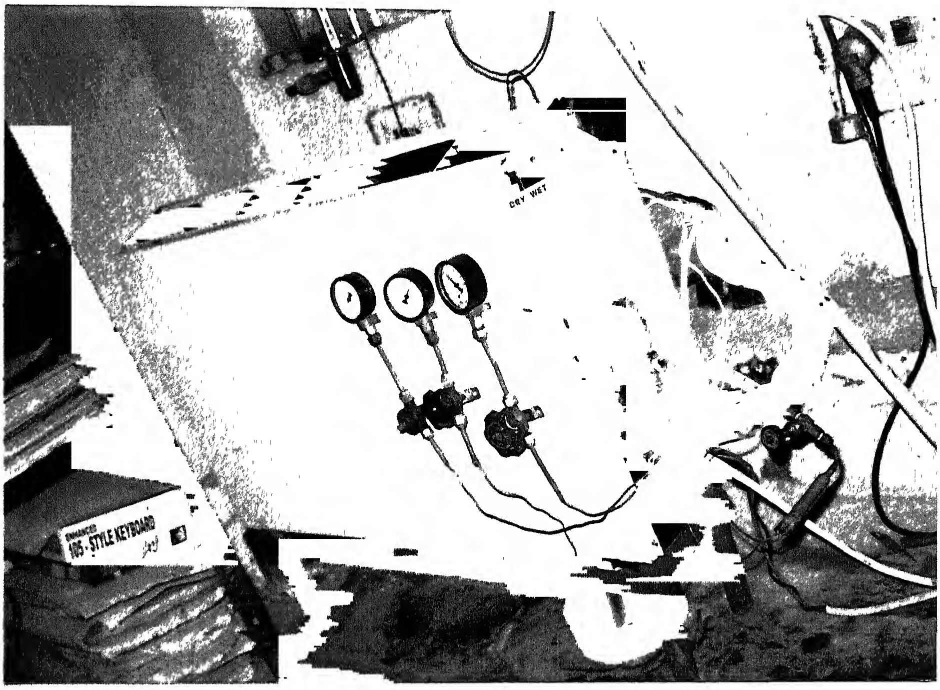


Photo 1. Evaporator unit

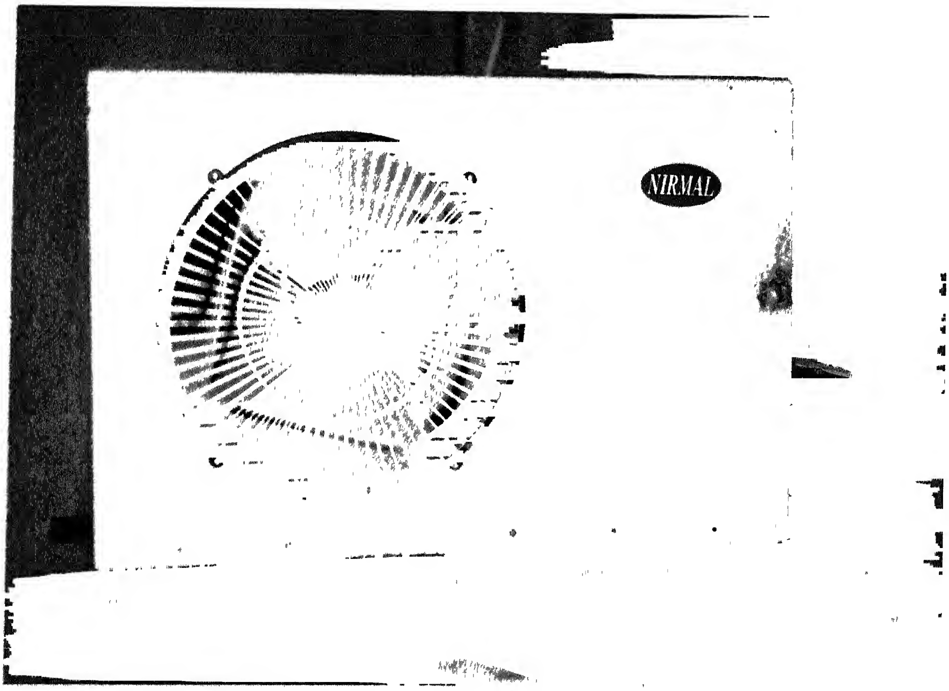


Photo 2. Condenser unit

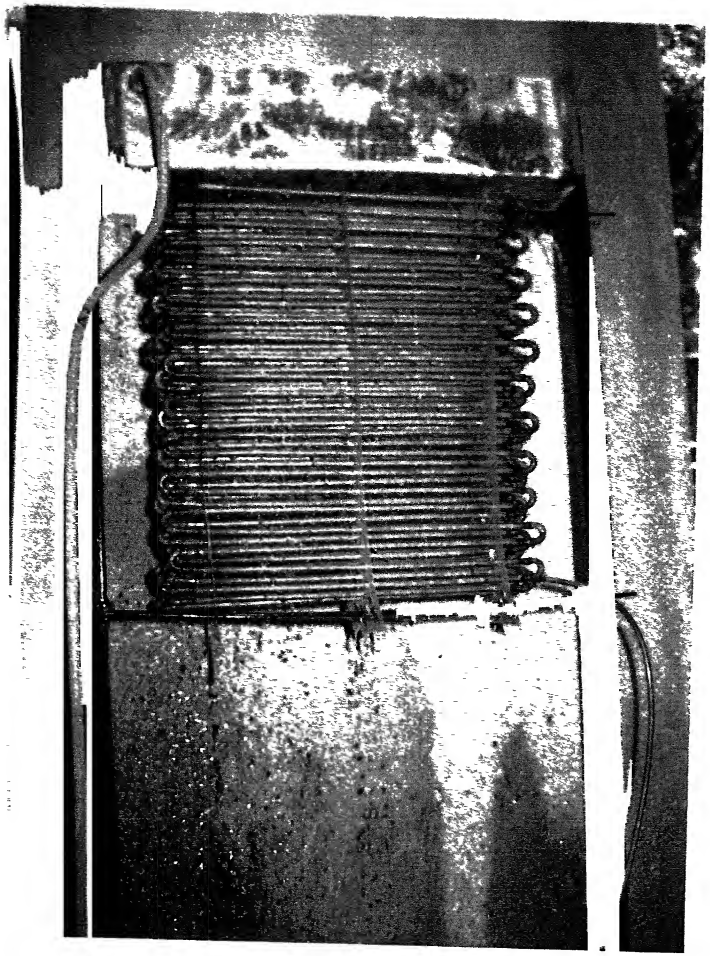


Photo 3. Evaporative condenser

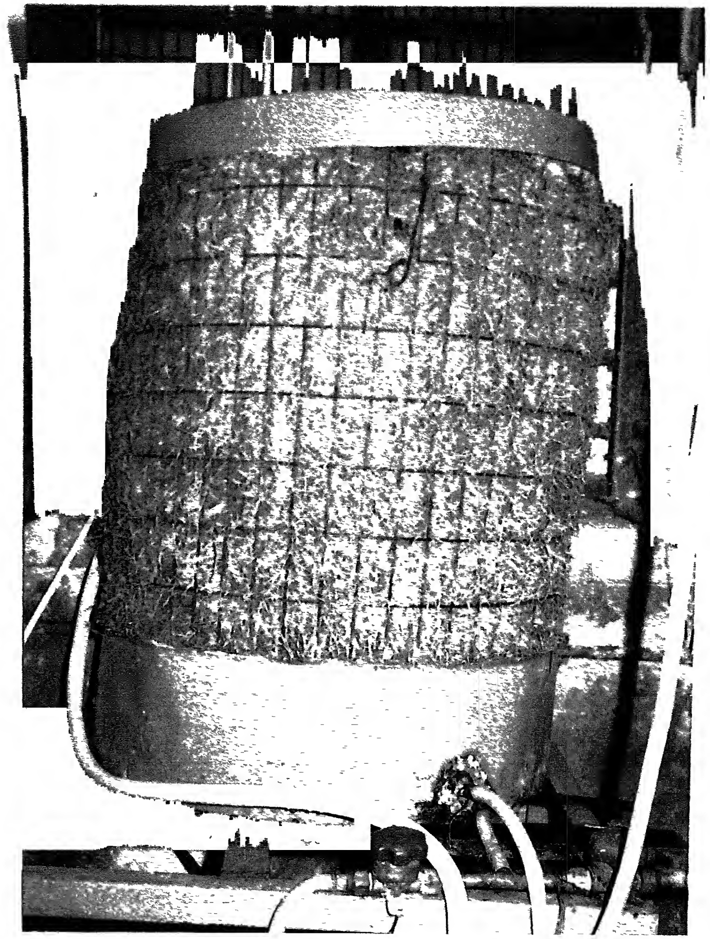


Photo 4. Desert cooler

5.2 Instrumentation

In the present study the following instruments are used

- Pressure gauges
- Watt meter
- Ammeter
- Rotameter
- Anemometer

Three pressure gauges are fitted in the system to measure the pressures at outlet of the condenser, at inlet and outlet of the compressor. Current take by the compressor is measured by ammeter and the power consumption of the system is measured by a single phase, two-wire wattmeter. A rotameter of 5.0 USGPM is used to measure the flow rate of the water while running the desert cooler. A vane type anemometer is used to measure the air flow rate at different points through the outlet of the evaporator when split air conditioner is operating in different modes and also at the outlet of the desert cooler.

5.3 Capillary design

A program was developed to calculate the increment in length of the capillary for different pressure drops and the same is used when evaporative condenser is used in the system. The capillary replaced is of 11 inches in length and 1.7 mm inner diameter, two in number. Hand shut-off valves are provided after condenser and before the evaporator to facilitate for changing the capillary of different lengths for testing.

5.4 Leak Check

Once all the brazing work and final installation is completed the system is tested for leakage. The system is filled with Nitrogen to a pressure of 20 bar, for leakage in the system all the joints are tested with soap solution and then it is left for 48 hours. Other method of leak testing is by using electronic leak detector. Then the system is evacuated by a vacuum pump to a vacuum pressure of 25 inches of Hg. The vacuum test is also carried for 48 hours. Thus, finally it is ensured that the system is ready to carryout experiments.

5.5 Measurement of Flow Rate

The local or point velocity of a stream in a flow line is different at different points over the cross-section of the line, being a maximum at the centre and falling off towards the wall.

In order to find the flow rate, one needs to know the average velocity of the stream, that is, the velocity which, on being multiplied by the cross-sectional area of the pipeline and the density of the flowing fluid will give the quantity of fluid passing through the line per unit time.

The average velocity of a stream is measured as follows. The cross-section of the duct is divided into a number, n, of equal areas and the local velocity is measured at a representative point in each. This technique is known as *traversing for average velocity.*

On designating the velocity in each area as $v_1, v_2, v_3, \ldots\ldots, v_n$ (m/s), the cross-sectional area of the pipeline at exit as A (m^2), and the mass flow rate as Q_M (Kg/s), we get

$$Q_M = \rho A v_{av}$$

Where,

$$v_{av} = (v_1, v_2, v_3, \ldots\ldots, v_n)/n$$

Thus, volume flow rate is given by

$$Q_V = A v_{av}$$

5.6 Estimation of Running Cost

For the running cost of the system the main factor to be considered is electricity cost. In the present study the running cost is calculated on hourly basis taking into account the present cost of the electric charge per kWh. The running cost analysis is made for both the theoretical and experimentally calculated power consumption of the system.

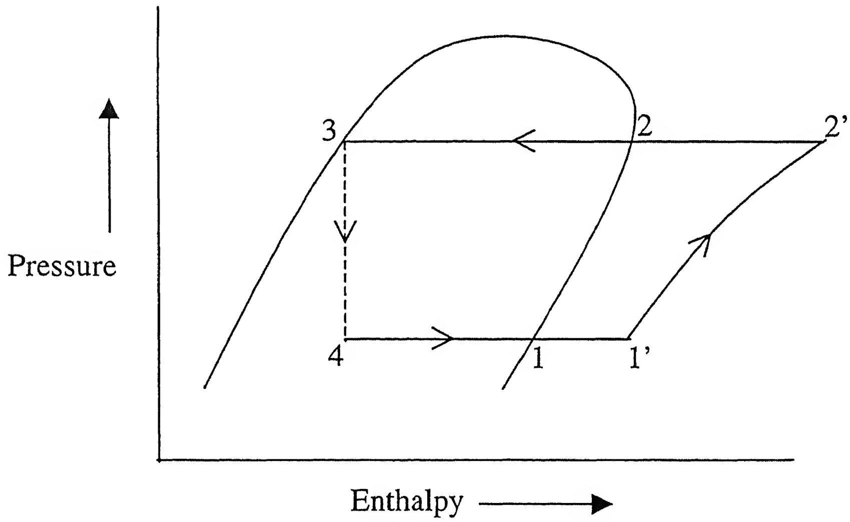


Figure 5.1 shows vapor compression cycle of the system

Power of the compressor is

$$P_c = \frac{m_{ref}\left(h_{2'} - h_{1'}\right)}{\eta_c} \quad \text{kW}$$

where $h_{1'}$ and $h_{2'}$ are enthalpies.

Power of desert cooler is

$$P_d = V \times I \times PF \quad \text{Kw}$$

where PF is the power factor and its value is taken as 0.86

The total running cost of the system includes SAC with conventional condenser, the SAC with the evaporative condenser and desert cooler.
The running cost of compressor C_1 is given by

$$C_1 = p_c \times C_E$$

The running cost of condenser fan is given by

$$C_2 = P_{fan} \times C_E$$

The running cost of evaporator fan C_3 is given by

$$C_3 = P_{fan} \times C_E$$

The running cost of pump C_4 is given by

$$C_4 = P_{pump} \times C_E$$

The total running cost of SAC with conventional air-cooled condenser T_1 is given by

$$T_1 = C_1 + C_2 + C_3$$

The total running cost of SAC using evaporative condenser T_2 is given by

$$T_2 = C_1 + C_3 + C_4$$

The total running cost of desert cooler T_3 is given by

$$T_3 = P_d \times C_E + C_4$$

The seasonal running cost of the hybrid air conditioning system is the sum of the running costs of Split Air-Conditioner and Desert cooler. Desert cooler is run

during hot and dry seasons for the months of April to June and October. The Split Air-Conditioner is run during the months July to September. Thus the accumulation of all the hours during which the system is operated gives the total running cost of the system.

Chapter 6

Results and Discussions

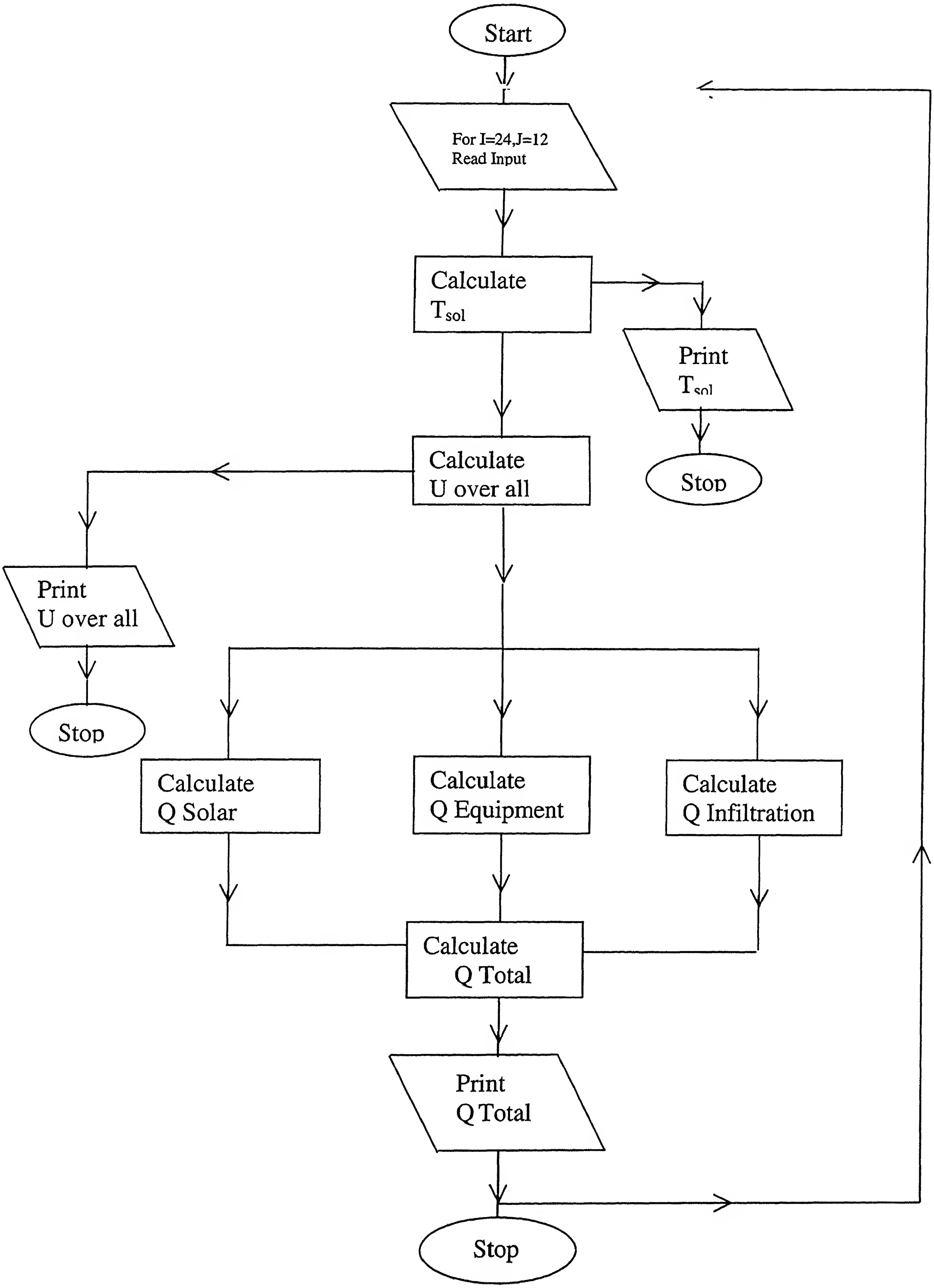
This section discusses in brief the results obtained in the present work. Cooling load calculations are made on the basis of the theoretical and actual temperature variation taking into consideration the effect of heat capacity and time-lag factor of structures. A generalized computer programme is made based on the mathematical formulation as explained in Chapter 3. Flow chart representing explicitly the cooling load calculations is presented in Fig 6.1.

6.1 Graphical Representation of Results

6.1.1 Theoretical and Actual Temperature Variation

The actual hourly temperature variation as well as the suggested sinusoidal hourly variation in temperature based on maximum and minimum temperature variation of the day has been shown in Fig 3.1. The actual temperature is generally seen to be slightly higher than the theoretical value from 8 A.M. to 2:30 P.M. just before peak hour. Thereafter, the actual temperature is lower than that of the theoretical value. The areas under both the curves reveal a small difference (3 to 8 %).

Flow Chart


Start
For I=24,J=12
Read Input
Calculate
T sol
Print
T sol
Stop
Calculate
U over all
Print
U over all
Stop
Calculate
Q Solar
Calculate
Q Equipment
Calculate
Q Infiltration
Calculate
Q Total
Print
Q Total
Stop

6.1.2 Sol-Air Temperature Variation for Different Walls and Roof

The ambient temperature and sol-air temperatures over the different walls and the roof are presented in Fig 6.2. The typical result presented is for 15th June. From this plot it is seen that the sol-air temperature for a horizontal surface (roof) is the highest and is maximum at noon, corresponding to the maximum incident solar radiation. From the plot it can be inferred that, the heat transfer through the east and west facing walls is more prominent than that of north and south facing walls. Thus the area of eastern and western walls should be less than that of northern and southern walls.

Fig 6.3 shows the variation of ambient temperature and sol-air temperature over different walls and the roof on 15th January.

6.1.3 Variation of Relative Humidity

Fig 6.4 shows the hourly variation of relative humidity for summer month on 15th June and for winter month on 15th January.

6.1.4 Solar Load

Fig 6.5 shows the daily average intensity of solar radiation throughout the year. The figure clearly shows the solar load variation and it is also shows the months where heating is required.

Fig 6.6 shows the variation of daily-averaged sol-air temperature for walls and roof throughout the year.

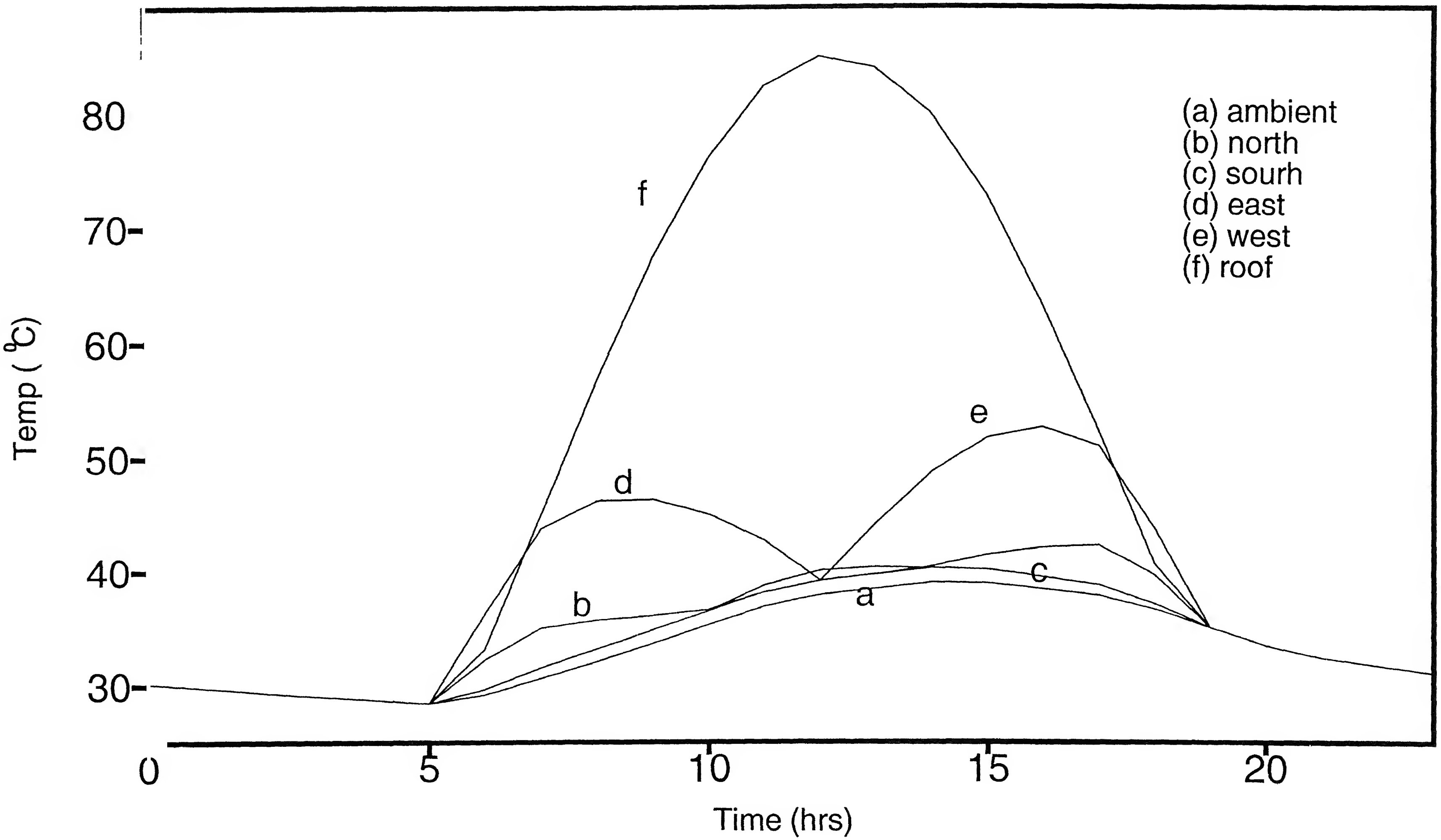


Figure 6.2 Variation of ambient temperature and sol-air temperatures for walls and roof on June15

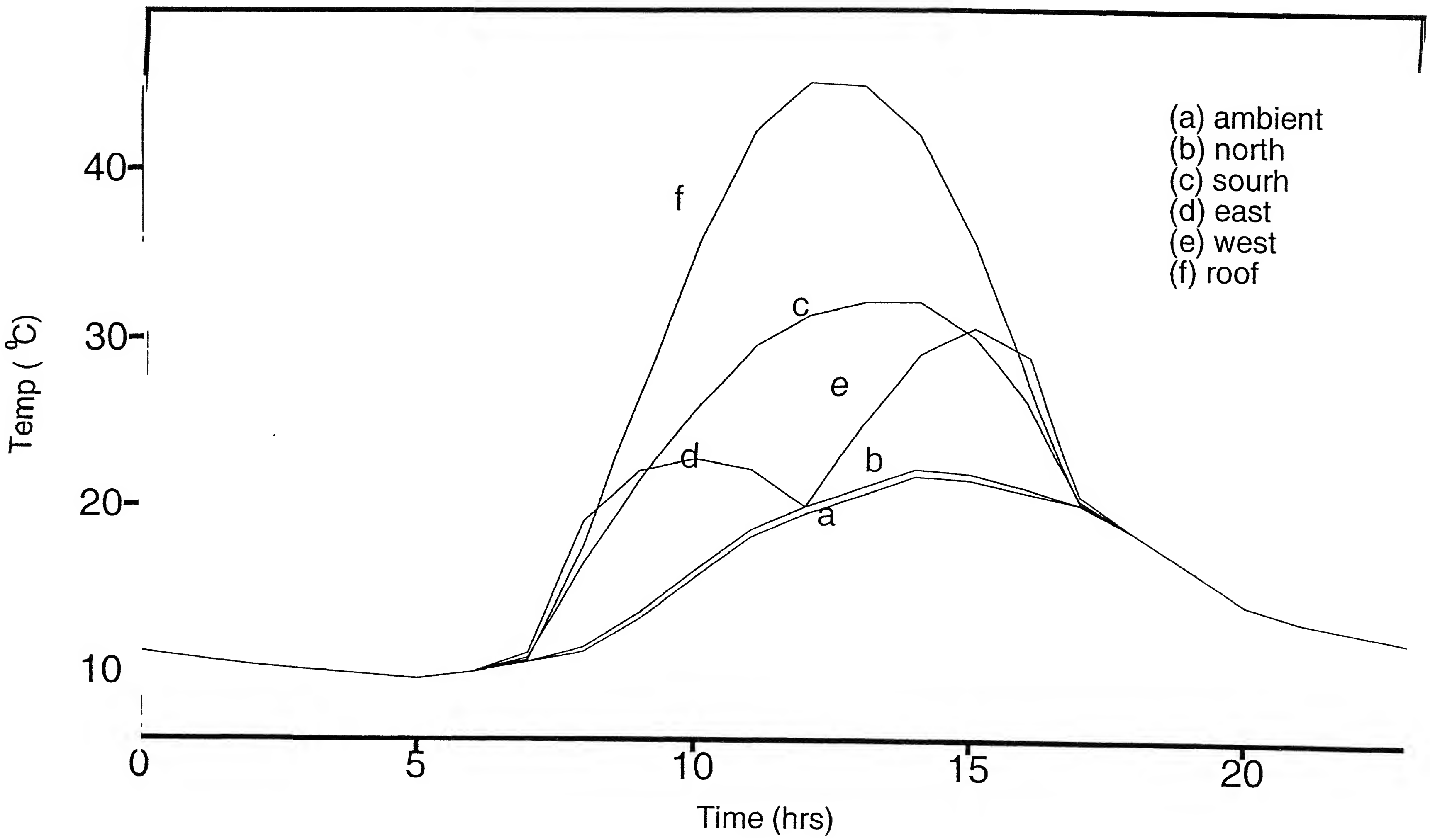


Figure 6.3 Variation of ambient temperature and sol-air temperatures for walls and roof on January15

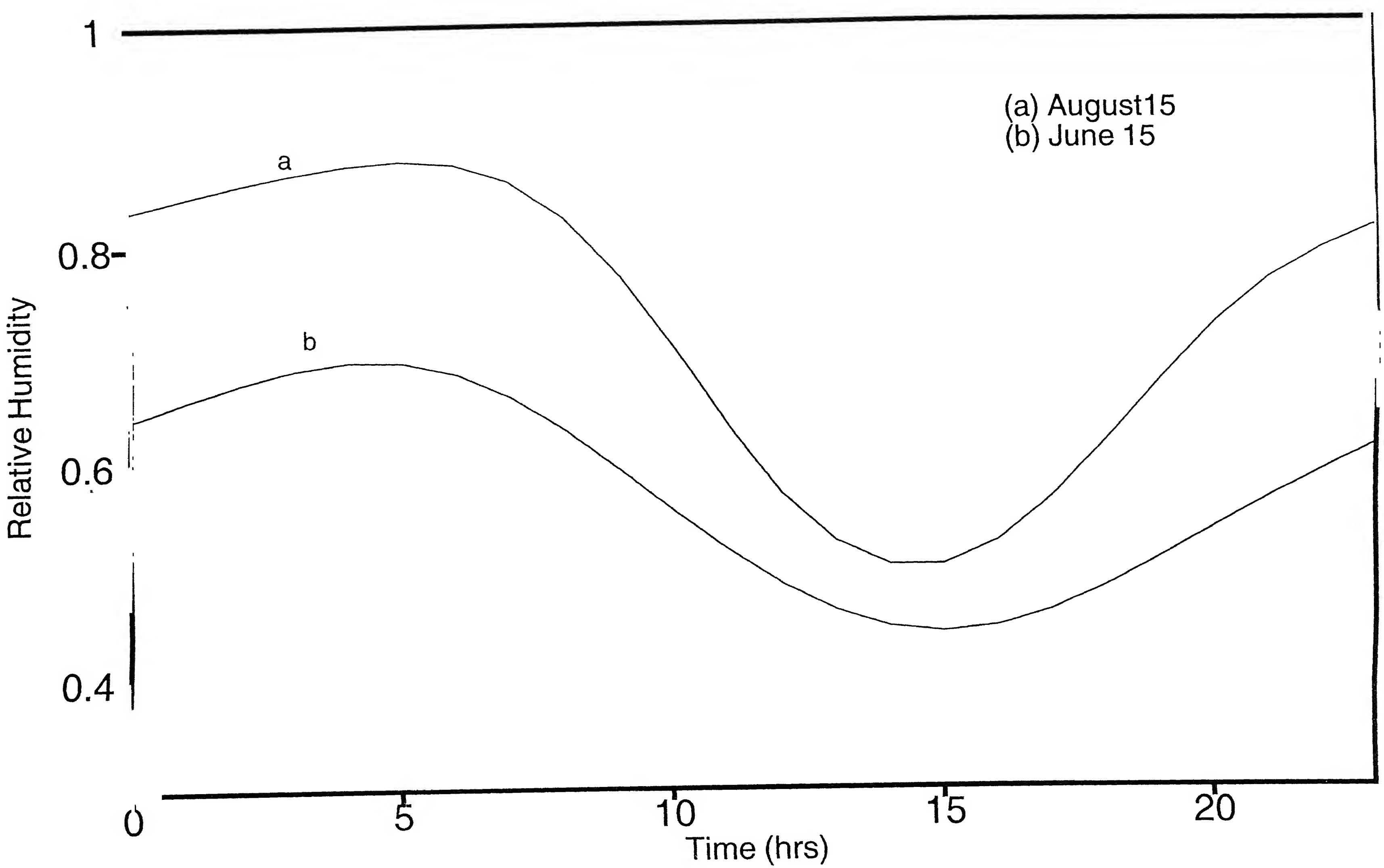


Figure 6.4 Hourly variation in Relative Humidity

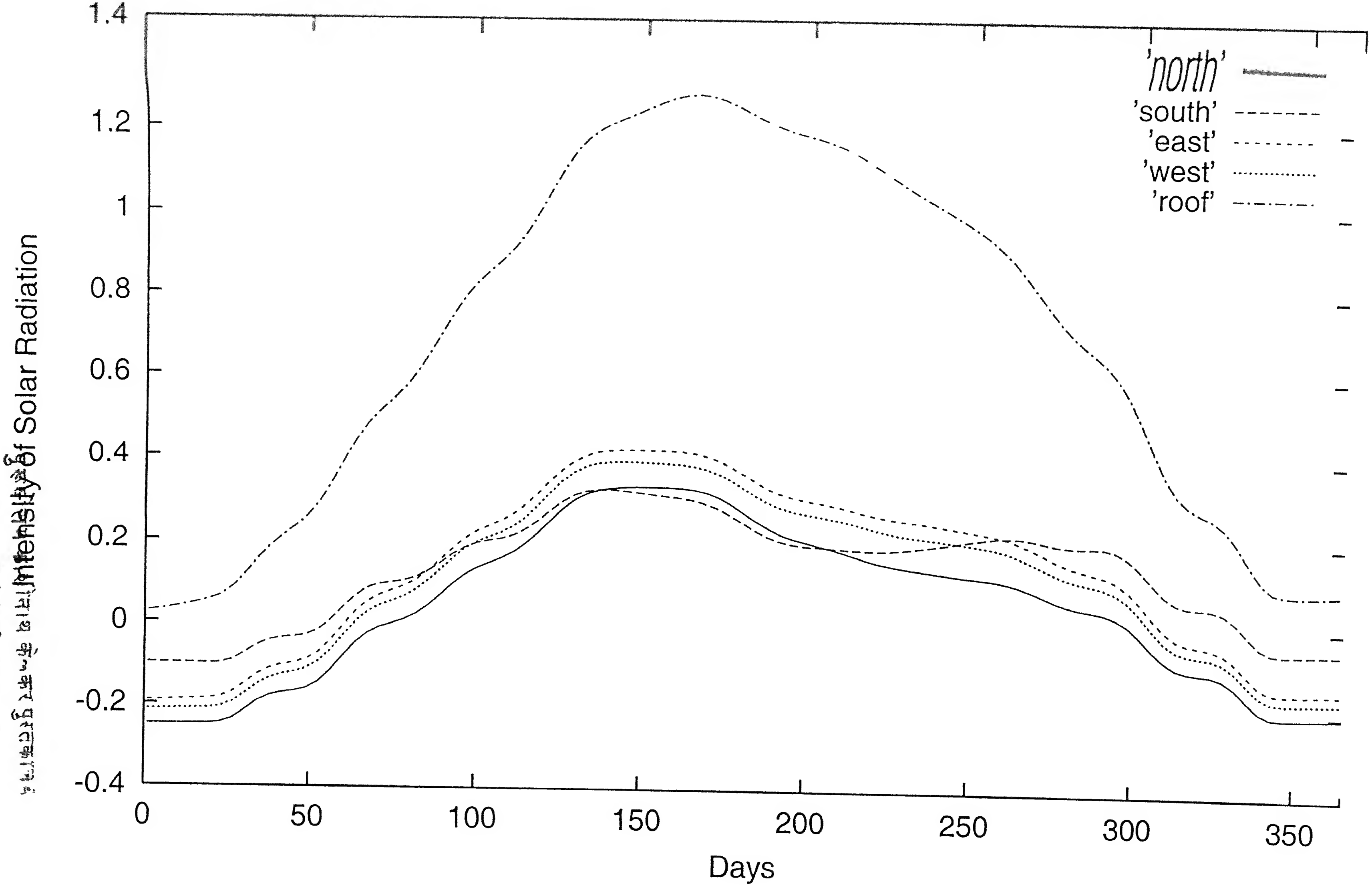


Figure 6.5 Variation of intensity of solar radiation for 365 days

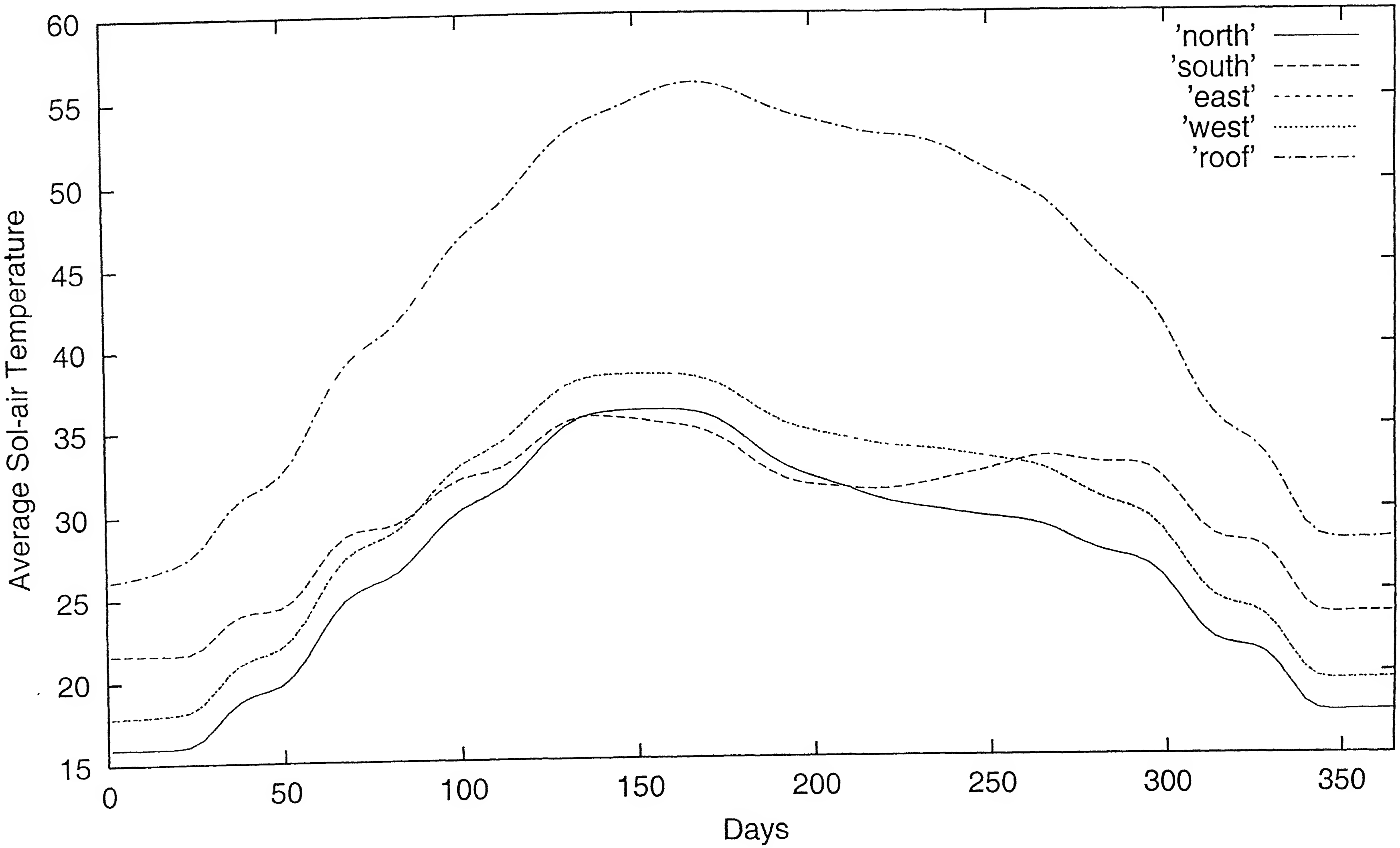


Figure 6.6 Variation of average sol-air temperature for 365 days

6.2 Experimental results

6.2.1 Graphical Representation of Experimental Results

Experiments are conducted in two modes one is with conventional split air conditioner having air-cooled condenser and other modified system using evaporative condenser. The refrigerating effect produced by the system is studied for various cooling water rates flowing through the evaporative condenser.

Figure 6.7 shows the condenser pressure that decreases with the increasing cooling water flowrate. Hence the head pressure for the condenser is reduced significantly, rendering better life of the compressor due to proper lubrication of the piston cylinder assembly.

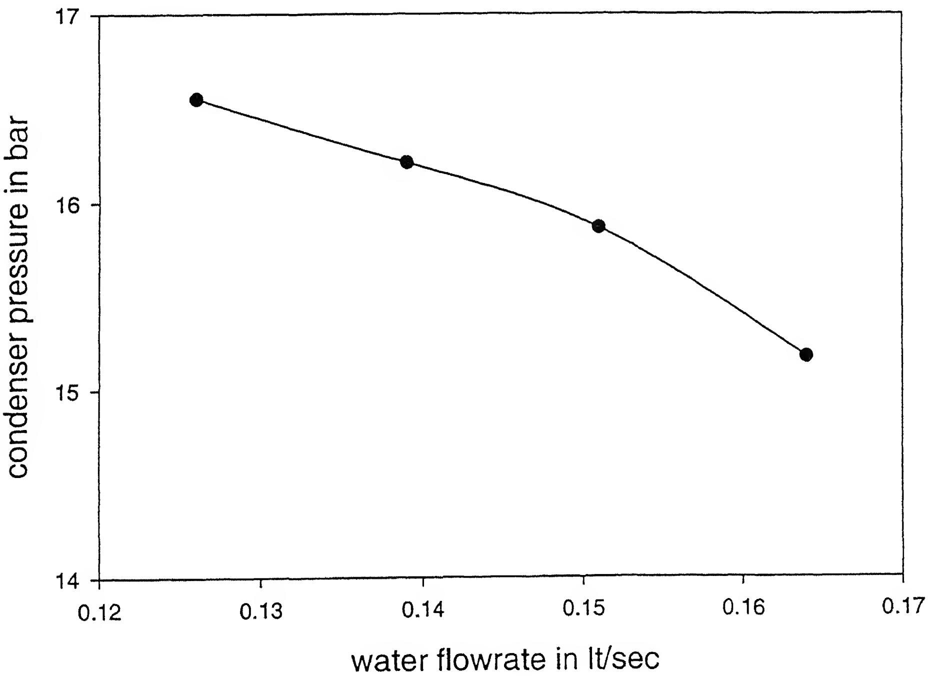


Figure 6.7 Variation of Condenser Pressure with Water Flowrate

Figures 6.8 and 6.9 show the variation of work of compressor as well as refrigerating effect with condenser pressure. Evidently higher the condenser pressure higher is the compressor work and lower is the refrigerating effect.

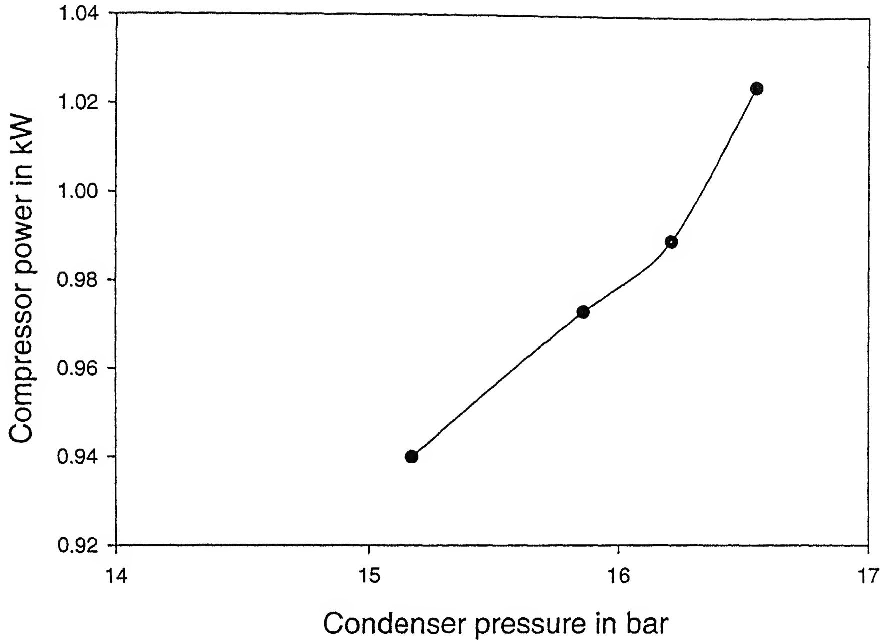


Figure 6.8 Variation of compressor work with condenser pressure

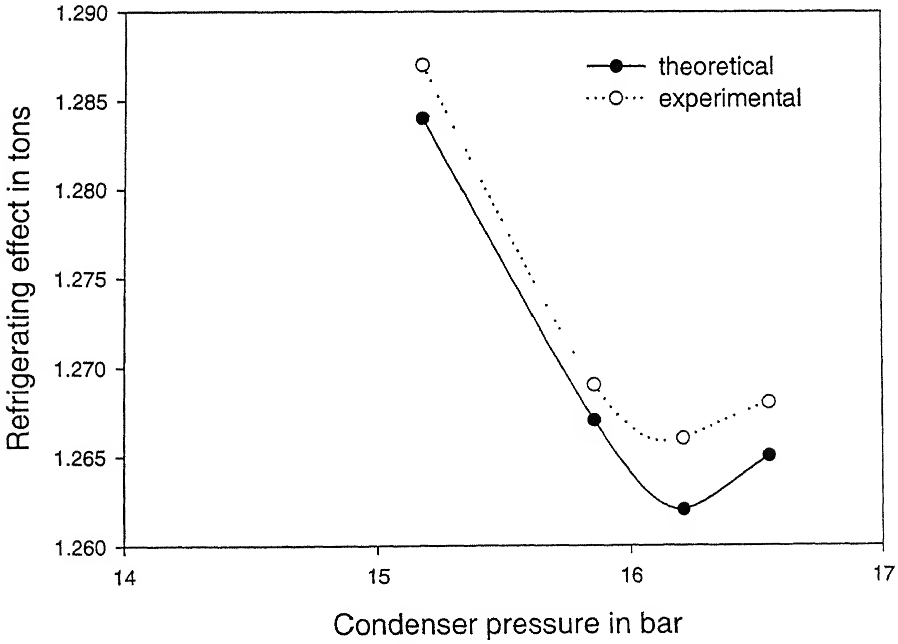


Figure 6.9 Variation of refrigeration effect with condenser pressure

6.3 Running cost

For the seasonal operation, i.e., for the months of April to October, running cost of the split air-conditioner with air-cooled condenser is Rs. 8560 and split air-conditioner with evaporative condenser is Rs. 7267. For the seasonal operation, running cost of the hybrid air-conditioner with air-cooled condenser is Rs. 6163 and hybrid air conditioner with evaporative condenser is Rs. 5379. Hybrid air-conditioner as compared to conventional SAC gives 28% (with air-cooled condenser) and 15.2% (with evaporative condenser) saving in running cost. Hybrid air-conditioner with evaporative condenser gives 12.7% saving in running cost than that of the hybrid air-conditioner with air-cooled condenser.

6.4 Tabulated Results

The experiments are conducted during September and October months and the results are tabulated. Table 6.1 shows experimental data for Desert Cooler on 14th September operated both in day mode and night mode. Tables 6.2 and Table 6.3 shows the experimental data for the Split Air-Conditioner with evaporative condenser and Split Air-Condenser with conventional air-cooled condenser on 1st October.

Experiments are conducted in the laboratory. The split air-conditioner with evaporative condenser has given 22.3% increases in the refrigerating effect than the split air-conditioner with air-cooled condenser. The energy consumption (per ton of refrigeration) of split air-conditioner with evaporative condenser is 15.1% less than that with air-cooled condenser. For seasonal operation the hybrid air-conditioner gives 38.9% energy saving than that of the air conditioner with air-cooled condenser and 56.3% as compared to the air conditioner with evaporative condenser.

Table 6.1 Tabulated values of the experimental data for Desert Cooler

Mode	Time	Air Conditions °C			Relative Humidity		Water flow-rate L/s	Energy meter reading kWh	Energy consumed for 1hr
		Inlet		Outlet					
		DBT	WBT	DBT	Inlet	Outlet			
Night mode	11.15	31.2	27.0	28.0	0.68	0.82	0.164	574.60	0.15
	11.30	31.2	27.0	28.0	0.69	0.83	0.164	574.63	
	11.45	31.5	27.5	28.4	0.70	0.83	0.164	574.66	
	12.00	32.0	27.5	28.2	0.73	0.85	0.164	574.72	
	12.15	32.5	28.0	28.4	0.69	0.85	0.164	574.75	
Day mode	12.30	32.5	28.0	28.5	0.64	0.84	0.164	574.75	0.17
	12.45	31.8	28.0	28.5	0.64	0.84	0.164	574.81	
	1.00	31.8	28.0	28.5	0.64	0.82	0.164	574.85	
	1.15	31.6	28.5	29.0	0.70	0.82	0.164	574.88	
	1.30	31.5	28.0	28.0	0.71	0.84	0.164	574.92	
No timer	1.45	32.0	27.5	28.0	0.72	0.86	0.164	574.10	0.34
	2.00	32.0	27.5	27.0	0.74	0.92	0.164	574.19	
	2.15	32.5	28.0	26.5	0.74	0.94	0.164	574.27	
	2.30	33.0	28.5	26.5	0.81	0.94	0.164	574.36	
	2.45	33.5	29.0	25.5	0.81	0.94	0.164	574.44	

Table 6.2 Tabulated values of the experimental data for SAC with evaporative condenser

SL .No	Flow rate Lt/sec	Air Conditions °C				Pressure Psi				Amount of condensate (ml)	Energy meter reading kW	Ammeter reading in amps
		Evaporator										
		Inlet		outlet		Comp.	Con.	Evaporator				
		DBT	RH	DBT	RH	Pre.	Pre.	Pre.	Temp.			
1	0.164	30.1	0.84	3.9	0.64	240	220	65	37°F	320	2.4	8.3
2	0.151	29.6	0.68	3.8	0.64	250	230	65	37°F	320	2.4	8.3
3	0.139	29.0	0.68	3.4	0.62	255	235	65	37°F	370	2.6	8.5
4	0.126	28.8	0.68	2.9	0.62	260	240	66	38°F	350	2.7	8.5

Table 6.3 Tabulated Values of the Experimental Data for SAC with Conventional Air-cooled Condenser

SL .No	Time	Air Conditions °C				Pressure Psi				Amount of Condensate (ml)	Energy meter reading kW	Ammeter reading in amps
		Evaporator										
		Inlet		outlet		Comp.	Con.	Evaporator				
		DBT	WBT	DBT	WBT	Pre.	Pre.	Pre.	Temp.			
1	3.05	26.6	22.5	12.2	10.5	302	298	62.5	38°F	370	2.4	10
2	3.25	26.1	22.0	11.7	10.5	301	292	61.0	37°F	380	2.4	10
3	3.45	26.1	22.0	10.5	9.7	300	290	60.0	34°F	270	2.1	10
4	4.05	26.2	22.0	10.5	9.7	300	290	60.0	34°F	245	2.1	10
5	4.25	26.0	21.5	9.6	8.9	300	290	60.0	33.5°F	265	2.1	10
6	4.45	25.8	21.0	9.2	8.9	300	290	60.0	33.5°F	265	2.0	10

Chapter 7

Conclusions and Suggestions

7.1 Conclusions

1. A general program has been developed for calculation of the total cooling load on the basis of hourly temperature variation using T_{max} and T_{min} for a given day. The decrement factor and time lag have been used to account for the wall heat capacity and effect of unsteady nature of heat transfer. The overall heat transfer coefficient has been calculated by considering the dependence of inside convective heat transfer on the inside surface temperature and air motion. Newton-Raphson method has been used to calculate the same.
2. The energy consumption (per ton of refrigeration) of split air-conditioner with evaporative condenser is 15.1% less than that of the air-cooled condenser and 22.3% increases in the refrigerating effect than the split air-conditioner with air-cooled condenser.
3. The head pressure for the split air-conditioner with evaporative condenser is 40% less than that of conventional split air-conditioner. This will render better life of the compressor.
4. For seasonal operation, hybrid air-conditioner gives 38.9% energy saving than that of the air conditioner with air-cooled condenser. On the other hand 56.3% energy saving is obtained when the split air-conditioner uses an evaporative condenser.

5. Saving in running cost of hybrid air conditioner with evaporative condenser is 12.7% than hybrid air conditioner with air–cooled condenser. While that of hybrid air conditioner as compared to conventional SAC is 28% (with air-cooled condenser) and 15.2% (with evaporative condenser).
6. In the perspective of energy conservation and sound control the use of split air conditioner with evaporative condenser is a better choice.

7.2 Scope of Future Work

1. Experimental studies can be conducted to find the power consumption and refrigerating effect produced by the system for different capillary tube sizes by operating the system for whole of a season to arrive at exact practical data.
2. Economic analysis can be done for the varying electricity cost, including the equipment cost.

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Appendices

Appendix A

Temperature and Relative Humidity of Outside Air

Table A.1: Average temperature and relative humidity of outside air at Kanpur, January to June.

Time	January		February		March		April		May		June	
	T	ϕ	T	ϕ	T	ϕ	T	ϕ	T	ϕ	T	ϕ
hrs	° C	%	° C	%	° C	%	° C	%	° C	%	° C	%
2:30	10.28	86.2	12.9	81.45	18.65	65 5	22.9	52.8	28.05	52.25	29.25	68.5
5:30	9.5	88.9	11.4	85.3	16.65	73.0	20.8	59.4	26.05	58.8	28.3	74.0
8:30	11.35	87.0	14.7	79.75	21.1	59.0	27.8	41.4	32.8	41.4	32.65	63.1
11:30	18.75	58.5	23.0	46.0	29.1	35.3	35.15	23.8	39.2	28.4	37.65	48.5
14:30	22.05	45.15	25.7	35.3	33.65	26.3	37.25	20.0	41.5	22.7	39.25	40.25
17:30	19.9	55	23.85	41.9	30.65	30.35	35.3	23.75	39.6	27.25	37.6	44.3
20:30	13.65	77.1	17.15	67.3	23.95	50.45	28.55	38.55	32.6	41.3	32.8	56.5
23:30	11.5	83.15	14.7	76.5	20.95	63 0	25 35	45.95	29.78	48.0	30.4	64.0

Table A.2: Average temperature and relative humidity of outside air at Kanpur, July to December.

Time	July		August		September		October		November		December	
	T	ϕ	T	ϕ	T	ϕ	T	ϕ	T	ϕ	T	ϕ
hrs	^oC	%	^oC	%	^oC	%	^oC	%	^oC	%	^oC	%
2 30	27.5	90.0	26 85	93.55	25.25	89.45	20.3	85 7	14 3	84.65	10.7	85.85
5 30	26 95	91.95	26 55	93.5	24.3	91.8	18.95	88 9	12.7	89.3	9.2	88.7
8·30	29 0	84 55	28.2	89.55	27.51	72.35	24.3	72.9	17.75	79.85	13.1	82 0
11·30	32.0	72 4	30.8	80.15	31.25	65.75	31.8	44.15	27.2	41.35	22.4	43.1
14:30	33 2	68.25	31.2	76.3	32.5	60.3	33.2	40 05	29.3	32 65	24.65	34 35
17:30	32.45	71.85	30 0	81 25	30.5	68.68	28.55	57.15	23.3	59.15	20.3	57.3
20 30	29 6	81.85	28.15	89	27.25	82.6	23.5	74.05	18.2	76.4	13.3	77.0
23 30	28 4	86.1	27 35	91.5	26.05	86.7	21.05	81.75	15.85	82.5	12 05	82 65

Table A.3: Maximum and minimum temperature of outside air at Kanpur.

Month	Maximum Temperature	Minimum Temperature
	^oC	^oC
January	23.1	8.0
February	26.5	10.05
March	32.65	15.4
April	38.15	19.9
May	41.9	25.18
June	41.35	27.6
July	34.55	26.25
August	32.62	26.04
September	33.6	23.85
October	34.36	18.23
November	30.15	11.75
December	26.05	8.1

Appendix B

Values for Solar Intensity Calculation

month	Equation of Time Minute	Declination Angle Degree	A W/m^2	B	C
Jan	-11.2	-20.00	1230	0.142	0.058
Feb.	-13.9	-10.80	1214	0.144	0.600
March	-7.50	0.00	1185	0.156	0.071
April	1.10	11.60	1135	0.180	0.097
May	3.30	20.00	1103	0.196	0.121
June	-1.40	23.45	1088	0.205	0.134
July	-6.20	20.60	1085	0.207	0.136
August	-2.40	12.30	1107	0.201	0.122
Sept.	7.50	0.00	1151	0.177	0.092
Oct.	15.40	-10.60	1192	0.160	0.073
Nov.	13.80	-19.80	1220	0.149	0.063
Dec.	1.60	-23.45	1233	0.142	0.057

A = Apparent solar irradiation at air mass = 0, W/m^2
B = Atmospheric extinction coefficient
C = Diffuse radiation factor

Appendix C

Particulars of the Room for Cooling Load Calculations

Room Size	: 4.87 x 4.57 x 4.27m^3
Area of East Facing Wall	: 4.27 x 4.57 m^2
Area of West Facing Wall	: 4.27 x 4.57 m^2
Area of North Facing Wall	: 4.87 x 4.27 m^2
Area of South Facing Wall	: 4.87 x 4.27 m^2
Area of Roof	: 4.87 x 4.57 m^2
Thickness of Wall	: 0.25999 m
Thickness of Roof	: 0.25999 m
Number of Fans	: 1
Rating of Fans	: 60 Watt
Number of Lights	: 3
Rating of Lights	: 40 Watt

Appendix D

Properties of Moist Air

$$Ps1 := \frac{221.287}{\left[e^{\left[\left[7.4379+\left(1.152\,10^{-5}-4.787\,10^{-9}\cdot Tdb1\right)\cdot(Tdb1-483.15)^{2}\right]\cdot\left(\frac{647.3}{Tdb1}-1\right)\right]}\right]}$$

$$Pv1 := \varphi 1\cdot Ps1$$

$$\omega 1 := 0.662\cdot\left(\frac{Pv1}{P - Pv1}\right)$$

$$h1 := 1.004\cdot Tdb1 + \omega 1\cdot(2501.4 + 1.884\cdot Tdb1)$$

Appendix E

Correlations for the Properties of R22

$$hf := 46.2102 + 1.20394(t) + 6.8280\left(\frac{t}{100}\right)^2 - 8.99387\left(\frac{t}{100}\right)^3 + 20.0612\left(\frac{t}{100}\right)^4$$

$$hg := 251.106 + 0.354577(t) - 19.4993\left(\frac{t}{100}\right)^2 - 7.62005\left(\frac{t}{100}\right)^3 - 11.6756\left(\frac{t}{100}\right)^4$$

$$sf := 0.181132 + 0.0043667(t) - 0.0596756\left(\frac{t}{100}\right)^2 - 0.0234343\left(\frac{t}{100}\right)^3 + 0.0638047\left(\frac{t}{100}\right)^4$$

$$p := 4.98105 + 0.161746 + 19.5655\left(\frac{t}{100}\right)^2 + 9.8434\left(\frac{t}{100}\right)^3 + 2.22194\left(\frac{t}{100}\right)^4$$

$$vg := X - Y + 3.4673222\frac{(E)}{(t + 273.15)} - 0.0043008\frac{(F)}{p} + 0.0624118G$$

where

$$X := 0.0077416R\cdot\frac{(t + 273.15)}{p} - 0.2795307(C)\cdot\frac{p}{\left[\frac{(t + 273.15)}{100}\right]^2}$$

$$Y := 0.0173324(D)\cdot\frac{p^{0.4}}{(t + 273.15)^4}$$

$R := 0.123937$ $C := 0.00493$ $D := 4.75$ $E := 0.3985$ $F := 3.54$ $G := 0.0561$

$$vgs := (1 + 0.004636\,tsup)\,vg$$

$$\eta_c := \frac{1}{\left[1.13623 + 1.13289\left(\frac{rp}{10}\right) - 3.34529\left(\frac{rp}{10}\right)^2 + 4.8675\left(\frac{rp}{10}\right)^3 - 2.135\left(\frac{rp}{10}\right)^4\right]}$$

$$cp := 0.6 + 0.0005246\left[(t + 45) + 0.043686(t + 45)^2\right] \qquad \text{for} \quad -60 \le t \le 10 \text{deg C}$$

and

$$cp := 0.70114 + 0.0029529\left[(t - 283.15) + 0.010472(t - 283.15)^2\right] \qquad \text{for} \quad 10 \le t \le 60 \text{deg C}$$